A COMPARITIVE ASSESSMENT OF NAVAL SHIP PROPULSION OPERABILITY --HIGH PERFORMANCE vs. CONVENTIONAL DESIGNS

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PROPULSION SYSTEM OPERABILITY--

HIGH PERFORMANCE VS. CONVENTIONAL DESIGNS

by

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ABSTRACT

High performance ships in general, are faster and more maneuverable than displacement ships of comparable size, while achieving parity in payload-carrying capability. This performance results from the design and implementation of low impact subsystems which allow the high performance ship to absorb the cost, in space and weight, of increased horsepower and installation of a lift system, the two major factors which contribute to the speed and seakeeping advantage. By designing displacement ship systems to high performance standards, an improvement in payload-carrying capability, or some other performance area, can be realized. The propulsion system is one system which offers great potential for space and weight savings. Ship design is compromise and any improvement of one feature dictates a degradation of another feature. In reducing propulsion system impact, system operability has been sacrificed. There are many features of high performance and conventional displacement ships and their propulsion systems which influence operability. The differences in these features can be identified and analyzed to determine the degradation of high performance propulsion system operability to achieve low weight and volume impact. Once these features and their influence on operability are known, they can be judiciously used by the ship designer to minimize the degradation of conventional propulsion system operability while still reducing the impact of the system in terms of weight and volume.

Thesis Supervisor: Clark Graham
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NOMENCLATURE

C_P	Specific heat for constant pressure, $BTU/lb_m{}^\circ R$
C_V	Specific heat for constant volume, $BTU/lb_m{}^\circ R$
g_o	Gravitational constant, 32.17 ft/sec^2
h_o	Stagnation entholpy, BTU/lb_m
J	Mechanical equivalent of heat, 778 ft lb_f/BTU
P_o	Stagnation pressure, psi
SHP	Shaft horsepower
T_o	Stagnation temperature, °R
Δ	Full load displacement of ship, tons
∇	Total enclosed volume of ship, cubic feet
γ	C_P/C_V
ρ	Gas density
M_n	Number of men assigned to a functional category, where n is a subscript defining the category
V_n	Volume of a functional category
W_n	Weight of a functional category
HRS	Total number of working hours per week of the propulsion division
HRS_m	Number of hours dedicated to a function by the propulsion division per week, where m defines the function

<u>Definition of subscripts (n)</u>

E	Propulsion division
ES	Engineering shops
MACH	Propulsion machinery

MP	Main propulsion system
MS	Main machinery space
R	Repair division
STRM	Storerooms
2	SWBS Group 2 Propulsion plant
230	Propulsion units
240	Transmission and propulsor systems
241	Reduction gears
242	Clutches and couplings
243	Shafting
244	Shaft bearings
245	Propulsors
246	Propulsor shrouds and ducts
247	Waterjet propulsors
250	Propulsion support systems (except fuel and lube oil)
260	Propulsion support systems (fuel and lube oil)
298	Propulsion plant operating fluids
299	Repair parts and special tools

Definition of subscripts (m)

CM	Corrective maintenance
FM	Facility maintenance
PM	Preventive maintenance
UW	Underway
WS	Watchstanding

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CHAPTER 1

INTRODUCTION

High performance ships are capable of higher speeds and exhibit superior seakeeping characteristics than their conventional displacement counterparts, without sacrificing payload carrying capability. Higher speeds are achieved in part with the use of low impact propulsion systems. The reduced weight and volume of such systems allow high performance ships to carry more installed shaft horsepower relative to their size and thus increase speed.

Superior seakeeping is achieved by decoupling the influence of the sea on the hull. This decoupling is brought about with the use of a lift system such as foils or air cushions. The lift system also contributes to reducing the drag of high performance ships, allowing them to reach higher speeds.

These improvements in ship performance do have an impact on the overall ship design. The added weight and volume required by the lift system penalizes the entire design. Yet even with the penalty of the lift system, high performance ships have maintained a payload carrying capability at least as good and often better than conventional displacement ships. To accomplish this, other functional areas have been designed for low ship impact by making use of low specific weight components.

The ability of high performance ships to carry more payload by making use of low impact systems is an attractive characteristic. It has been proposed that conventional displacement ships use the same high performance design standards to reduce the impact of various systems and improve their ability to carry payload. Studies by Grostick[1] and Fahy[2] have evaluated high performance design standards and shown the feasibility of applying them to conventional displacement ships.

High performance ships are by nature weight limited with respect to their lift system capability. Very often the feasibility of a design depends on the designer's ability to keep the weight of each functional area within its allocated budget. To do this, trade-offs must be made. It has been suggested the operability of systems has been sacrificed to meet budgets and insure design feasibility.

Operability is a summation of many characteristics which measure a system's ability to perform its required mission subject to varying conditions and constraints. This includes the ease of operation and maintenance by the crew, the system flexibility, vulnerability and reliability.

If operability has indeed been sacrificed in high performance ships, it may still be possible to take advantage of high performance design standards in conventional displacement ship systems, since conventional ships are usually

more conservatism in their design. This conservation may allow the use of high performance design standards with only minimal affects on operability.

A few functional categories offer the greatest potential for weight and volume savings on conventional displacement ships. The propulsion system is one of these. There is a potential 40% savings in weight and 25% savings in volume by designing a conventional displacement ship's propulsion system to high performance standards. The resulting affect on system operability is still to be determined.

This analysis will investigate the impact high performance design standards have on propulsion system operability. Appropriate high performance and conventional ships will be selected and a brief comparison of design standards conducted to show the potential for weight and volume savings. An appropriate measure of operability will be presented along with its relationship to various features of ship and propulsion system design. Next, differences in high performance and conventional displacement propulsion systems will be assessed for their effect on operability. Finally, the operability predictions made by the ship designers will be compared to determine how well these predictions are supported by ship and propulsion system characteristics.

CHAPTER 2

THE IMPACT OF HIGH PERFORMANCE DESIGN ON WEIGHT AND VOLUME

The purpose of this chapter is to present a summary comparative analysis of high performance and conventional displacement ships. The superior payload carrying capability of high performance ships will be demonstrated along with the penalty absorbed in the form of a lift system.

Various indices will measure the impact of high performance and conventional design standards and show how systems may be designed for reduced ship impact. The weight and volume savings realized by use of high performance design standards can then be reallocated to increasing payload or some other performance feature of the ship.

2.1 Selection of Ships

In order to conduct a meaningfull impact study of high performance technology, proper ships must be selected for comparative analysis. Ship pairs must be selected which meet the following guidelines:

--modern design

--fully combatant

--sufficient design data available

--similar in size and mission capability

In addition, there are variations in functional impacts due to size variations, so it is of value to look at both small ships and large ships.

2.1.1 Small Ships

The U.S. Variant of the NATO hydrofoil (PHM) is selected as the small high performance ship. It is a 245 ton, single-mission-area, gas turbine powered ship with a small crew, high speed, and limited endurance. The PHM is scheduled to be operational in the U.S. fleet in 1977. It is constructed of aluminum and is capable of speeds in excess of 40 knots.

The displacement counterpart selected is the PG-84 class patrol gunboat, operational since 1966, but with similar mission capability. It is the Navy's first combatant ship with gas turbine propulsion and aluminum hull construction. The PG-84 displaces 242 tons and is capable of calm water speeds of about 40 knots. Table 1 lists the general characteristics of both ships.

2.1.2 Large Ships

There is no current large hydrofoil available for study. There is, however, a 3000 ton Surface Effect Ship (LSES) in contract design for delivery to the fleet in the early 1980's. It is designed to perform destroyer type missions, has about a 3000 nautical mile range and is capable of speeds in excess of 40 knots. For its size, it has a much smaller crew size than current destroyers. The LSES is gas turbine powered and uses four water jet propulsors for mobility. Another two gas turbines provide power to lift

TABLE 1

SMALL SHIP CHARACTERISTICS

	PG-84	PHM-1
Displacement (TONS)	242	245
Length (FT)	165	146
Beam (FT)	24	24.4
Draft (FT)	5	6
Main Engines	(1) LM1500 G.T. (2) DIESELS 14,750 SHP CODOG	(1) LM2500 G.T. (2) DIESELS 17,340 SHP CODOG
Propulsor	(2) CRP Propellers	(3) Waterjet pumps
Electric Plant	(2) Diesel 60 HZ 100 KW	(2) G.T. 400 HZ 200 KW
Max Speed (KTS)	~40	+40
Range (N.M.)(est)	~500@ 40 KTS	~700 @ +40 KTS
Complement	29	21
Payload	3"/50 cal gun Standard missile or 40mm gun MK 87 FCS	76mm OTO Melera gun Harpoon missile MK 94 FCS

fans. The LSES is constructed of aluminum.

The conventional displacement ship selected for comparison with LSES is the FFG-7, a new class of guided missile frigate. The FFG-7 displaces 3600 tons and has a maximum speed of under 30 knots. Its hull is constructed of steel while its superstructure is aluminum.

Although, FFG-7 and LSES are ideal for comparison in size and mission capability, it is noted that the LSES has double the installed propulsive shaft horsepower of the FFG-7. If lift systems are considered, LSES has triple the installed shaft horsepower. The LSES propulsion system is similar to that of the DD-963 class destroyers. It is of interest to use the DD-963 in the comparison as well because of this similarity.

The DD-963 is a 7750 ton destroyer capable of speeds in excess of 30 knots. Its hull is constructed of steel and its superstructure of aluminum. The DD-963 has a longer range than the FFG-7 and LSES but has similar mission capabilities. The general characteristics of these ships are given in Table 2.

2.2 Functional Classifications

There are many different methods used in classifying the functional areas of ships, among them the Bureau of Ships Consolidated Index (BSCI) and the Ship Work Breakdown

TABLE 2

LARGE SHIP CHARACTERISTICS

	DD-963	FFG-7	LSES
Displacement (TONS)	7747	3605	3000
Length (FT)	563	445	266
Beam (FT)	55	45	108
Draft (FT)	18	15	19
Main Engines	(4) LM2500 G.T. 86,000 SHP	(2) LM2500 G.T. 41,000 SHP	(4) LM2500 G.T. 86,000 SHP
Propulsor	(2) CRP Propellers	(1) CRP Propeller	(4) Waterjet Pumps
	(3) G.T. 60 HZ 2000 KW	(4) Diesel 60 HZ 1000 KW	(3) G.T. 60 HZ 375 KW (3) G.T. 400 HZ 375 KW
Max Speed (KTS)	31+	28+	40+
Range (N.M.)	~6000 @ 20 KTS	~4500 @ 20 KTS	~3000 @ 20 KTS
Complement	260	176	97
Payload	(2) 5"/54 guns (1) MK 28/4 ASROC (2) Lamps helo (2) MK 32 TT (1) 20mm CIWS (1) MK 25/1	(1) 76mm OTO Melara (1) MK 13 (2) Lamps helo (2) MK 32 TT (1) 20mm CIWS	Standard missile Harpoon missile (2) Lamps helo (2) MK 32 TT (2) 20mm CIWS

Structure (SWBS). Before the establishment of these standard classification systems, each ship designer used his own method of classifying weight and volume, manning, and energy allocation. When conducting an impact study, a consistant and concise classification system is mandatory in order to compare different ships on a common base. The system used in this analysis incorporates a functional breakdown that provides the necessary information to determine the relative impact and importance of the various ship functions.

The ships will be broken down into several functional categories:

Payload

Personnel

Ship Operations

Mobility

Electrical

Auxiliary

Hull Structure

Lift

Other Ship Systems

A detailed breakdown of each of these functional groups is given in the following sections.

2.2.1 Payload

Payload is made up of all items related to communications, detection, weapons, and miscellaneous payload. This includes command and control systems, exterior communications, surface and underwater surveillance systems, electronic countermeasures, launching and fire control systems, ammunition handling systems, magazines and ammunition, aircraft, and special mission facilities.

2.2.2 Personnel

The functional area of personnel includes the specific areas of crew living, personnel support, and personnel stowage. It emcompasses everything required for dealing with the human presence aboard ship.

2.2.3 Ship Operations

Ship operations includes ship control systems, maintenance, and tankage. Such elements as navigation, telephone systems, fire extinguishing systems, damage control, deck auxiliaries, shops and maintenance areas, ballast tanks, peak tanks, voids and unassigned spaces make up the functional area of ship operations.

2.2.4 Mobility

The main propulsion system and endurance fuel make up the major part of mobility. Also included are propulsion plant foundations, machinery space ventilation, feed water and lubricating oil.

2.2.5 Electrical

The electrical area consists of electrical power generation and support systems, and associated foundations.

2.2.6 Auxiliary

Climate control systems, sea water systems, fresh water systems, air and gas systems are all part of auxiliary.

2.2.7 Hull Structure

There is no volume associated with hull structure. The weights include shell plating and support structure, hull structural bulkheads, decks, platforms and flats, deckhouse structure, masts, kingposts and free flooding liquids.

2.2.8 Lift

This category includes all elements associated with foils or air cushions, lift fans, hydraulic systems, flexible seals and skirts.

2.2.9 Other Ship Systems

This functional area is made up of power distribution systems, lighting systems, underway replenishment systems, stowage spaces, special systems and miscellaneous liquids, along with any items not specifically included in any of the other functional areas.

2.3 Development of Parameters

Four general parameters or design indices are useful in comparing ship design standards. They are:

Functional Allocations

Specific Ratios

Densities

Capacity-Ship Size Ratios

Functional allocations are the weight, volume, and manning fractions of a functional category; for example, the function weight divided by full load displacement. They provide an indication of the relative impact or priority that subsystems or functional areas have on the total ship.

A functional category may have a small functional allocation because of low ship impact or because another category may dominate, thus driving the relative impact of the first category down. For this reason, there is a need to identify absolute ship impact. Specific ratios are suited to this task.

A specific ratio is the ratio of the "cost" of a function to its capacity. This normalizes results in a more meaningful comparison of functional impacts between ships and the impact of high performance technology can be more readily assessed.

Density is the weight of a function divided by its volume. Functional densities are useful in assessing whether a ship is weight or volume limited and what features are causing the limitation.

A capacity-ship size ratio normalizes a functional capacity relative to ship size. It is an indication of the functional importance and the emphasis the function places on the design.

These indices will be used to compare the impact of high performance design standards on the ship and its systems.

2.4 High Performance vs. Conventional Displacement Ships

Figures 1 through 4 are graphic representations of the weight and volume allocations computed for each of the five ships. They readily display the impact of the lift systems on the high performance ships. In the case of the LSES, the low lift system weight is deceiving. There is actually a penalty in additional fuel to operate the lift gas turbines.

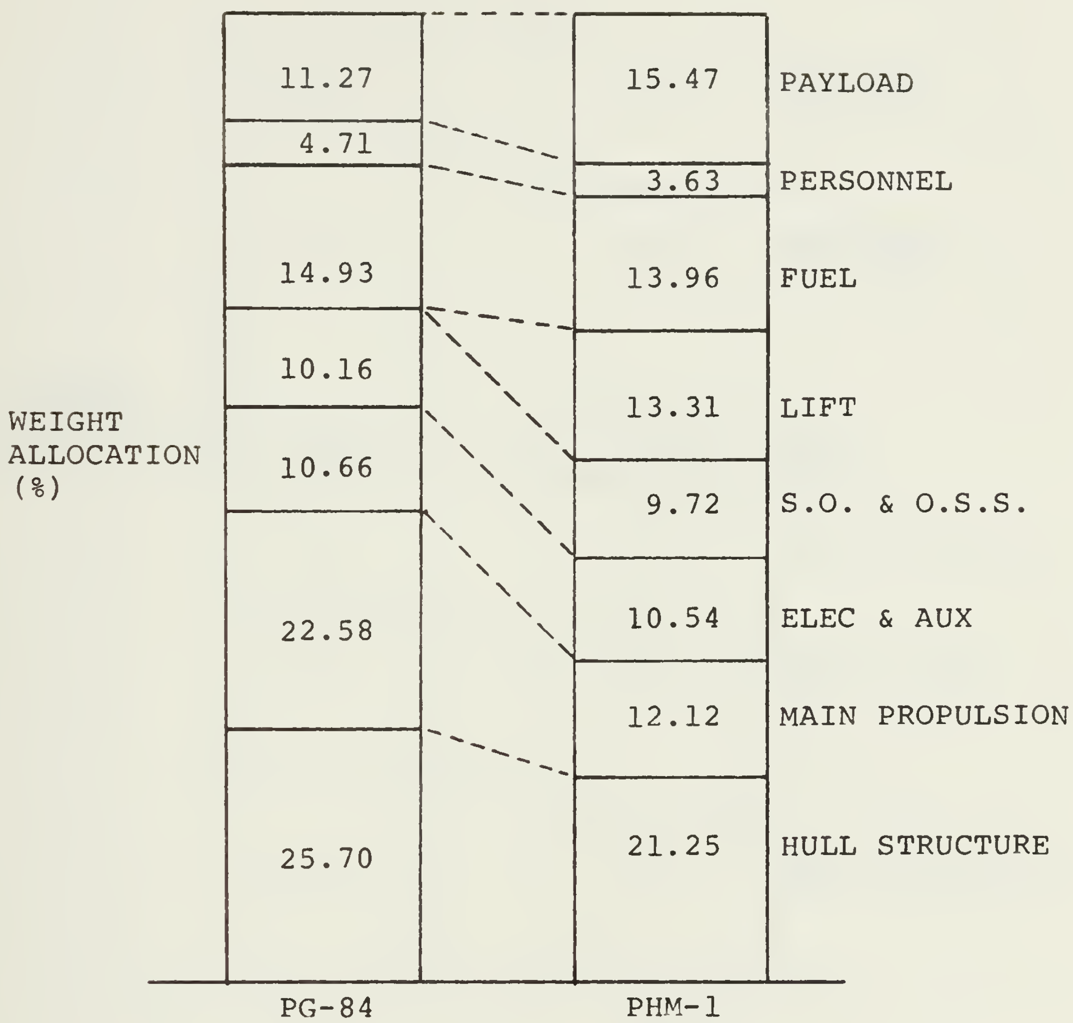

FIGURE 1 - COMPARISON OF WEIGHT ALLOCATIONS - SMALL SHIPS

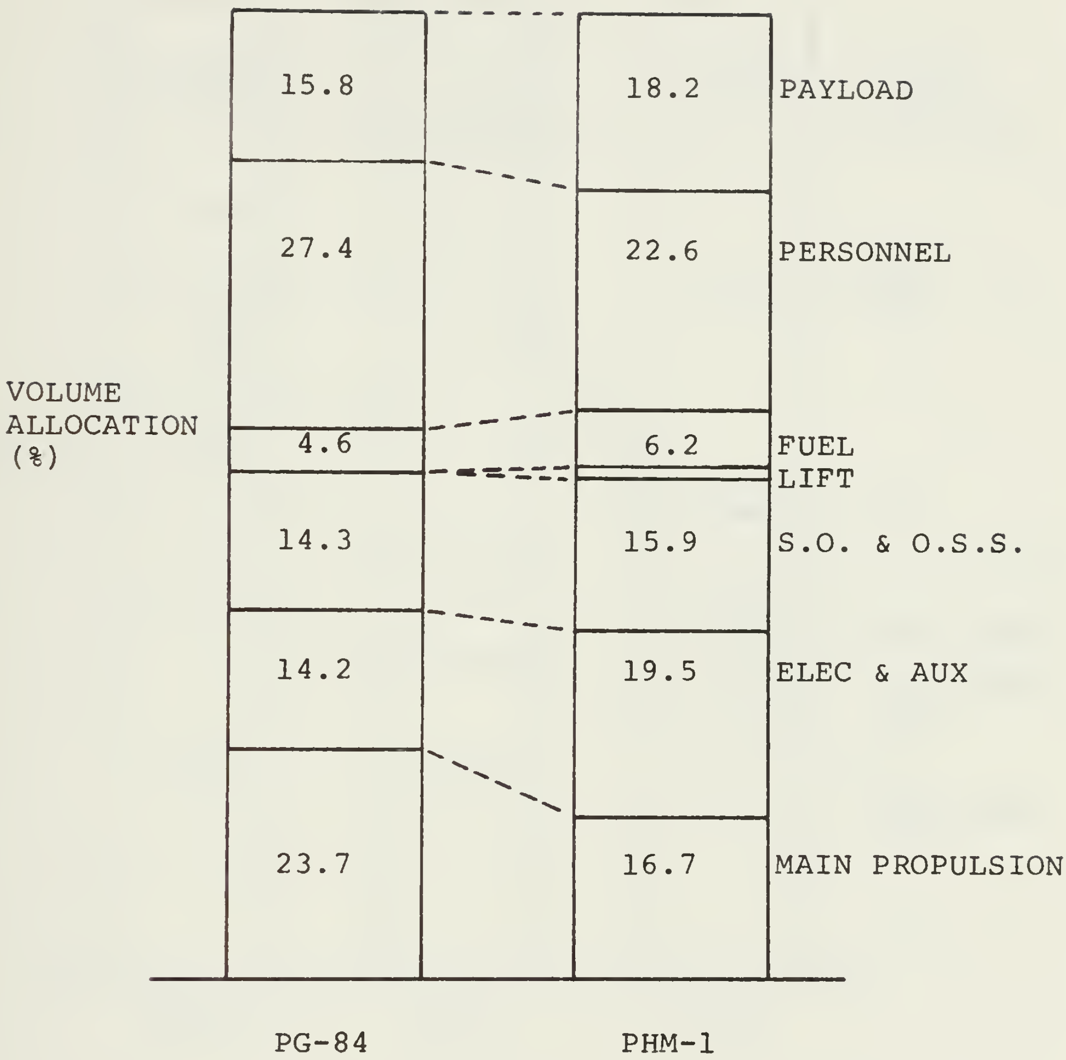

FIGURE 2 - COMPARISON OF VOLUME ALLOCATIONS - SMALL SHIPS

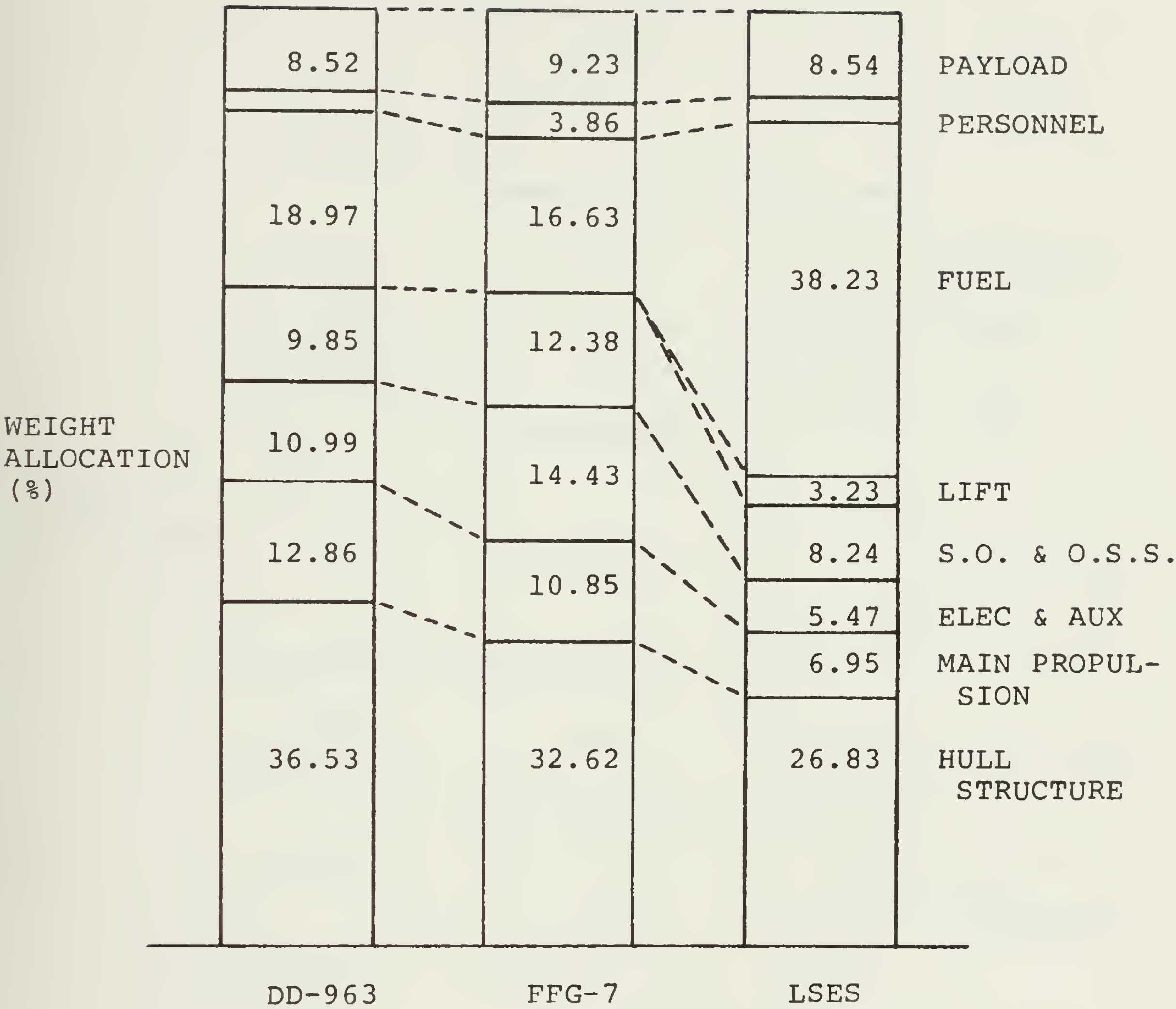

FIGURE 3 - COMPARISON OF WEIGHT ALLOCATIONS - LARGE SHIPS

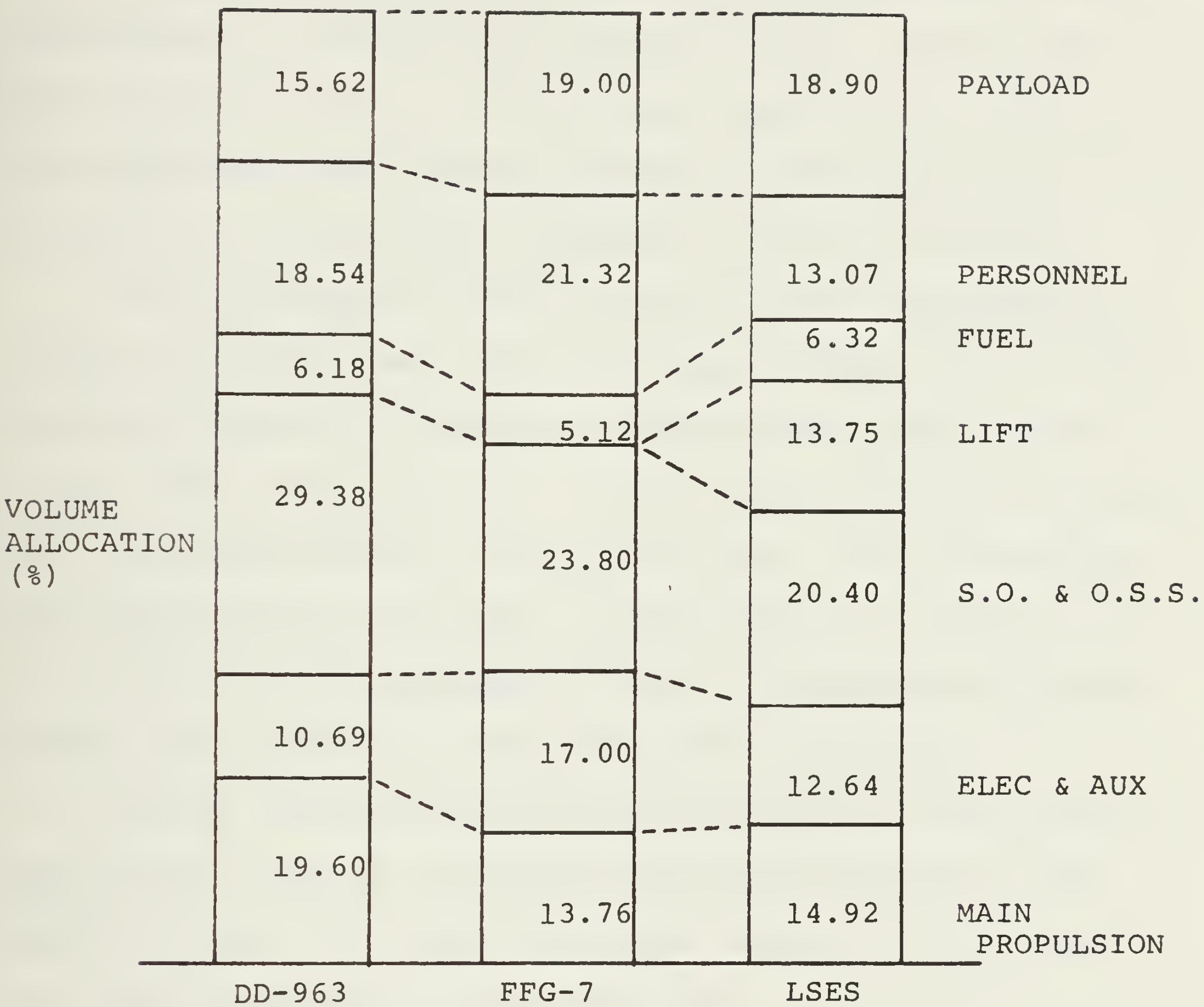

FIGURE 4 - COMPARISON OF VOLUME ALLOCATIONS - LARGE SHIPS

As shown in the figures, one-eighth of PHM's full load displacement is required for the foil system, yet its payload weight fraction is still higher than PG-84's. This is accomplished by low relative impacts of other functional categories, primarily main propulsion and hull structure.

The LSES demands 41.46% of its full load displacement to meet its lift, speed, and range requirements; but in spite of these impacts, has the same payload weight fraction as DD-963. The DD-963 and FFG-7 devote 18.97 and 16.63% of their full load displacement to speed and range. The high performance ship is then absorbing a weight penalty of nearly 25% of its full load displacement. This is accomplished through reduced ship impact of other functional categories.

Payload volume fractions are also larger in high performance ships, despite the volume they must allocate to lift systems. Again, the other functional areas are more compact and have less impact on the total ship.

High performance design standards can be used to reduce the weight and volume impact of a system without degrading its capacity. Thus, the savings can be reallocated to payload in the conventional displacement ship.

One functional area in particular offers a great deal of potential savings if designed to high performance standards. This functional area is the propulsion system, where a 40% weight and 25% volume savings are possible.

The impact of high performance design standards will be evaluated in the next section.

2.5 High Performance vs. Conventional Displacement Propulsion Systems

The propulsion system is an area where significant differences in design standards exist. Table 3 lists important propulsion characteristics and designs indices for each ship.

2.5.1 PG-84 vs. PHM

The first major differences between the two ships is found in the main propulsion system weight and volume allocations. PHM has significantly lower weight and volume fractions than PG-84, while at the same time having greater maximum speed and more installed horsepower.

Looking at specific ratios:

	PG-84	PHM
W_{MP}/SHP	7.11 lb/SHP	3.52 lb/SHP
V_{MP}/SHP	0.78 ft^3/SHP	0.44 ft^3/SHP

PHM's propulsion specific weight is less than half that of PG-84's. This can only result from the use of a considerably lighter propulsion system. The specific volume is lower for the PHM, due to the compactness of the high performance design.

TABLE 3

PROPULSION SYSTEM CHARACTERISTICS AND DESIGN INDICES

Item	Units	PG-84	PHM	DD-963	FFG-7	LSES
W_{MP}/Δ	%	19.4	11.1	10.0	8.0	6.9
V_{MP}/∇	%	23.7	16.7	19.6	13.8	14.9
W_{MP}/SHP	lb/SHP	7.12	3.52	20.13	15.84	5.38
W_{230}/SHP	"	1.52	1.36	2.12	3.37	0.61
W_{240}/SHP	"	3.42	1.41	10.43	7.32	1.98
W_{241}/SHP	"	1.34	0.44	3.79	2.87	0.45
W_{242}/SHP	"	0.27	0.03	0.10	0.00	0.02
$W_{243-247}/SHP$	"	1.81	0.94	6.53	4.45	1.50
$W_{250-260}/SHP$	"	1.41	0.58	12.88	4.14	1.10
W_{298}/SHP	"	0.74	0.14	1.01	0.91	1.67
V_{MP}/SHP	ft^3/SHP	0.78	0.44	2.39	1.73	1.38
W_{230}	tons	10.0	10.5	81.3	61.7	23.6
W_{240}	"	22.5	10.9	400.3	133.9	75.9
W_{241}	"	8.8	3.4	145.6	52.5	17.4
W_{242}	"	1.8	0.2	3.9	0.0	0.8
$W_{243-247}$	"	11.9	7.3	250.8	81.4	57.7
$W_{250-260}$	"	9.3	4.5	494.4	75.7	42.4
W_{298}	"	4.9	1.1	38.8	16.6	64.0
SHP	SHP	14,750	17,340	86,000	41,000	86,000
M_{MP}	men	12	6	27	10	6
SHP/Δ	SHP/TON	60.98	70.77	11.10	11.37	28.67

To determine the reasons for the differences in propulsion specific weight, subsystem specific ratios are examined:

		PG-84	PHM
PRIMEMOVER	W_{230}/SHP	1.52 lb/SHP	1.36 lb/SHP
TRANSMISSION	W_{240}/SHP	3.42 lb/SHP	1.41 lb/SHP
SUPPORT SYS	$W_{250-260}$/SHP	1.41 lb/SHP	0.58 lb/SHP
OPERATING FLUIDS	W_{298}/SHP	0.74 lb/SHP	0.14 lb/SHP

Significant differences occur in all areas except the prime mover specific weight, which is comparable for the two ships since similar prime movers are used in both. PHM employs waterjet propulsion, which is the principal reason for the lower transmission specific weight. Support systems specific weight is lower because the waterjet propulsor and reduction gear require less support in the areas of propulsion control and lubrication.

Since the ships already use comparable prime movers, the subsystem that is now dominating the propulsion system is the transmission subsystem. Elements contributing to the transmission specific weight are:

		PG-84	PHM
REDUCTION GEAR	W_{241}/SHP	1.34 lb/SHP	0.44 lb/SHP
CLUTCHES, SHAFTING AND PROPULSOR	$W_{242-247}$/SHP	2.08 lb/SHP	0.97 lb/SHP

The PHM's reduction gear is considerably lighter because it is coupled to a waterjet pump which requires much less torque and a smaller RPM reduction than the controllable reversable pitch propellers employed by the PG-84. The waterjet pump and shafting also has a considerably lower weight impact than the shafting and propellers required by PG-84.

2.5.2 DD-963, FFG-7 vs. LSES

Although the FFG-7 and LSES are of equivalent size, the significant comparison is between DD-963 and LSES since these ships have similarly configured and rated propulsion systems.

Again there are major differences in weight fractions. The LSES has the lowest impact propulsion system, yet has significantly more shaft horsepower installed per ton full load displacement than the conventional designs.

The differences in propulsion system specific ratios are as follows:

	DD-963	FFG-7	LSES
W_{MP}/SHP	20.13 lb/SHP	15.84 lb/SHP	5.38 lb/SHP
V_{MP}/SHP	2.39 lb/SHP	1.73 lb/SHP	1.38 lb/SHP

To achieve the low specific weight, LSES must have a much lighter propulsion system. It is necessary to go to the next level of detail to determine the reasons for the differences in specific weight ratios.

		DD-963	FFG-7	LSES
PRIME MOVER	W_{230}/SHP	2.12	3.37	0.61
TRANSMISSION	W_{240}/SHP	10.43	7.32	1.98
SUPPORT SYS	$W_{250-260}$/SHP	12.88	4.14	1.10
OPERATING FLUIDS	W_{298}/SHP	1.01	0.91	1.67

There are significant differences in all areas. All three ships use the same gas turbine as their prime mover so it might be expected that the prime mover specific weight ratios would be comparable. The major reason they are not is that in the DD-963 and FFG-7, the gas turbines are mounted on special bed plates and enclosed in a module, both of which add considerable weight. The LSES gas turbines are installed in their own machinery compartment but no special bed plate or module is used.

The LSES employs waterjet pumps which accounts for the significantly lower transmission specific weight. The lower support system specific weight is due to the lower support requirement of the waterjet pumps and reduction gears as discussed for small ships.

Looking at the transmission subsystem in more detail, the following specific weights are evaluated:

		DD-963	FFG-7	LSES
REDUCTION GEAR	W_{241}/SHP	3.79 lb/SHP	2.87 lb/SHP	0.45 lb/SHP
CLUTCHES, SHAFTING, AND PROPULSOR	$W_{242-247}$/SHP	6.63 lb/SHP	4.45 lb/SHP	1.52 lb/SHP

The observations made for small ships hold here as well. The lower torque and smaller RPM reduction requirements of the waterjet propulsors allow the reduction gear of the LSES to be lighter. The waterjet pump and shafting is also considerably lighter than the CRP propeller and shafting required by the conventional ships.

2.6 Summary and Conclusions

Even though high performance ships are penalized in weight and volume by their lift systems, they have better payload carrying capability than their conventional displacement

counterparts. To accomplish this, other function areas have reduced ship impact. The ship impacts of high performance propulsion systems have been reduced while installing more shaft horsepower relative to ship size. This is done by using components with low specific weights.

Therefore, conventional propulsion systems can be redesigned to reduce their ship impact by designing and installing components with low specific weights and volumes. The reduction of system impact will probably be less than that achieved by high performance ships. It is impractical to install waterjet pump propulsors on relatively slow ships since these propulsors are not as efficient as propellers at low speeds.

Ship design is trade-off. Improvement in one area requires a sacrifice somewhere else. To achieve the weight and volume savings in the propulsion system, some other feature must be degraded. The remainder of this analysis will address the possible sacrifice in propulsion system operability to reduce system impact on the overall ship.

CHAPTER 3

OPERABILITY

The operability of a system is a summation of characteristics which measure the ability of that system to perform its required mission under varying conditions and constraints. Characteristics such as reliability, availability, flexibility, maintainability, survivability, vulnerability and habitability are used as a measure of system operability.

Reliability of a system is the ability to successfully perform its function without interruption, once begun. Availability is the degree to which the system is able to start performing its function at any random time. Flexibility is the ability to meet system functional requirements with a variety of configuration operating modes. Maintainability is the ability of a system to be restored to service within a given period of time. Survivability is the ability to withstand battle damage and still meet functional requirements. Vulnerability is the susceptibility of the system to damage or failure. Habitability is the degree to which men can function in and around a system while it is operating.

This study will investigate three of these areas in detail since they are very descriptive of a propulsion system's operability. Reliability, availability, and maintainability are the characteristics of interest. In naval ship design

these characteristics are of such importance that they are used to specify performance requirements.

3.1 Reliability/Availability/Maintainability

Reliability of a system is the probability that it will perform its function without interruption once undertaken. The availability of a system is the measure of the degree to which a system is able to start performing its function at the start of any random mission time. Maintainability is the probability that a system will be restored to operation within a specific time when prescribed maintenance is performed.

For naval combatants it is obvious that a high availability is desired to counter any threat whenever it occurs. Likewise, when engaging a threat or performing a mission a high reliability is essential to ensure successful completion.

Reliability is defined mathematically by:

$$R = e^{-t/MTBF}$$

where t is the time elapsed since the beginning of the mission and MTBF is the mean-time-between-failures of the component or system in question. MTBF is determined by statistical analysis of past failures.

Availability is defined by

$$A = \frac{MTBF}{MTBF+MTTR}$$

where MTTR is the mean-time-to-repair and is also determined statistically.

Many times a component is not repairable due to lack of parts, tools, or crew ability. Many major propulsion system components on naval ships are designated non-repairable at sea by the crew because it is not feasible for the ship to physically carry the resources necessary to facilitate repairs. Systems which must be available for combat are also used in normal operations. When a component is non-repairable and is continually operated, its availability can only be as good as its reliability. Thus, for non-repairable components, availability is defined as

$$A = R = e^{-t/MTBF}$$

Redundancy is the degree of duplication a system has to meet its functional requirements. Often components with poor reliability are installed with one or more backup units. These units can be on-line standby or off-line standby. On-line standby infers the components are operating at an idle condition but ready to take the load if the primary component

fails. These units are subject to failure while in this idle mode, so a better method is off-line standby. In the off-line standby mode, the component is not in operation and therefore not subject to failure. It is dependent on a switching system to sense a failure of the primary component and turn on the backup.

Mathematically, the reliability of these backup configurations are defined as follows:

ON-LINE STANDBY $$R = 1 - (1 - e^{-t/MTBF})^n$$

OFF-LINE STANDBY $$R = e^{-t/MTBF} \times \left[\sum_{i=0}^{n-1} \frac{(t/MTBF)^i}{i!}\right]$$

where n is the number of components installed in parallel.

In naval propulsion systems there are some components in off-line standby and some in on-line standby. For this analysis, the assumption has been made that redundant components are in off-line standby.

3.2 Factors Affecting R/M/A

By their definitions, reliability and non-repairable availability are functions of time, decreasing with increasing time. Repairable component availability is independent of time and remains a constant value. Reliability is also dependent on the mean-time-between-failure of a component.

A longer MTBF improves reliability for all intervals of time. Likewise, a shorter mean-time-to-repair increases component availability. The reliability and availability of a component or system are significantly affected by MTBF and MTTR. In turn, there are numerous factors which affect MTBF and MTTR. These influencing factors are listed and discussed in the following sections.

3.2.1 MTBF Factors

The MTBF of a component or system is dependent on component loading, design characteristics, operating environment, designed service life, and the amount of preventative maintenance performed.

3.2.1.1 - Component Loading

If a given component was operated at capacities well below its designed rating, it would be expected to have a somewhat longer MTBF than the same component operated at its maximum capacity. Also, components operated in excess of their maximum ratings would be expected to show a shorter MTBF.

Given two components of the same capacity constructed of the same material and using similar design techniques, the smaller would be expected to exhibit a shorter MTBF due to the increased stress that must be absorbed.

These points will be investigated in much more depth in Chapter 4.

3.2.1.2 - Design Characteristics

Of two equivalently rated components, the one which is designed and manufactured using advanced techniques and higher strength steels should exhibit the best MTBF. Likewise, a component designed specifically for an application should have a better MTBF than an off-the-shelf multi-application component.

Design characteristic improvements are often used in combination with the reduction in size of a component. From such a combination it is not always easy to predict the variation in MTBF.

3.2.1.3 - Environment

Components not specifically designed for or protected from vibration, extremes in temperature and humidity, and salt water should show a reduction in MTBF when subjected to such an environment.

3.2.1.4 - Preventive Maintenance

Preventive maintenance is a planned series of operational checks and servicing actions which evaluate the condition of a component or system and restore it to a

higher mode of operability. All this is accomplished prior to a component reaching a failed state and requiring corrective maintenance. The intent of preventive maintenance is to extend the MTBF of component or systems.

3.2.1.5 - Component Service Life

Mechanical component MTBF is not always constant. An early break-in period is characterized by high failure rates and short MTBF's. During the operational period, MTBF remains at a constant value. Near the end of a component's service life, it enters a wear-out period in which failure rates once again increase and MTBF's become very short. Once this stage is reached, it is often more practical to replace the component than continue to repair it. The break-in and wear-out periods are very short compared to the operational portion of a component's service life so it is often assumed that MTBF is constant throughout the service life of a component.

3.2.2 MTTR Factors

Normally MTTR accounts for the time required to do the prescribed maintenance and assumes that all necessary spare parts and tools are immediately available along with the proper maintenance personnel. In actuality, this is not always a valid assumption. Due to differences in the

availability of spare parts and tools, the capability of repair shops, the accessability of the component and the number and skill level of maintenance personnel, identical components may exhibit different MTTR's from one ship to another.

3.2.2.1 - Repair Part Availability

The MTTR of a component will be long if the repair parts needed to restore it to operating status are not carried on board the ship. The delay encountered while waiting for the parts to be obtained from its supply source must be charged to MTTR. A ship having little or no space and weight allocated for spare parts storage is vulnerable to unavailability of spare parts.

3.2.2.2 - Tools Availability

Many complex components require special tools to facilitate repair. If such tools are not carried on board a ship, the delay caused is assessed to MTTR. Weight and volume budgets again dictate the availability of tools.

3.2.2.3 - Machinery Accessability

Delays caused by poor access to failed components is chargable to MTTR. Poor accessability can be due to a high density of components in a machinery space, or to hazardous conditions related to other operating machinery.

3.2.2.4 - Repair Shop Facilities

Many times the repair of a component or system may require rework of a failed part or fabrication of a new part. Inadequate shop facilities limit the repair work that can be accomplished and lengthen MTTR.

3.2.2.5 - Maintenance Manpower

Repair of components may require special skill levels not on board the ship, or several men may be needed to work simultaneously on the repair. When the appropriate skills or number of men are not available, the repair will take much longer, thus increasing MTTR.

3.2.3 Configuration

Propulsion system configuration is a combination of the type of components selected to perform required functions and the redundancy of components incorporated. Both redundancy and type of components influence system reliability and availability. More reliable and available components will increase system reliability and availability. Addition of many back-up components for those which are least reliable and available also improves system reliability and availability.

3.3 Summary and Conclusions

Reliability, availability, and maintainability are three of the most important areas of operability relative to propulsion system and total ship system performance. These areas are influenced by many aspects of the ship design. Any assessment of propulsion system reliability and availability should account for the factors which influence MTBF and MTTR for the analysis to be considered an accurate operability assessment.

In the following chapters, many of the factors discussed here will be evaluated for high performance and conventional displacement ships to determine what differences exist in the operability of their propulsion systems.

CHAPTER 4

MEAN-TIME-BETWEEN-FAILURES

In various comparative naval architecture studies, it has often been assumed that for a given power rating, a large heavy component was inherently more reliable than a small light weight one. The primary reason for this was the more conservative design and lower stress levels of the larger component. It must be remembered, however, that the high stress levels in the small, light weight design could be compensated for by more advanced design techniques and high strength materials.

Conservatism in design has been a trend in conventional displacement ship design for many decades. With no major difficulty in designing a ship which was feasible, conventional displacement ship designers allocated high percentages of total ship weight and volume to propulsion systems.

High performance ships must meet very tight weight constraints even to be feasible, so there is no room to be conservative. Many systems must be cut back to reduce their overall ship impact. Components must be lighter, more compact, and designed to higher stress levels with smaller factors of safety.

With this in mind, this chapter will investigate the major question of what effects on reliability can be expected when a component must be made more compact. Also, it has

been observed that sometimes an identical component is used on several ships but its maximum output may vary for each ship for one reason or another. In such a case, how might the reliability be expected to vary from ship to ship?

To answer these and other questions, a few propulsion system components will be selected and after determining what indices or parameters are best used to describe changes in reliability; basic engineering theory will be employed to assess the effects of such changes as size, power, and RPM.

4.1 MTBF Parameters

In mechanical systems, a failure occurs either when a component's parts have worn beyond acceptable tolerances or there is a breakage of a part. The wear occurs at a rate related to load level and type of materials in contact. Breakage is also related to load level and material characteristics. A good measure of MTBF then is the loading of the component or more specifically the stresses.

The components of interest to be analyzed are the gas turbines, the reduction gear, and the propulsors. In the case of the gas turbines, the expected operating life is governed by the amount of blade elongation. Due to high stress levels in the blades, accompanied by high temperatures, the blades will stretch with time by a phenomenon called

thermal creep. The creep rate is highly dependent on temperature. Thus, stress and temperature are key elements in assessing gas turbine reliability. It should be noted that there are other contributing factors to gas turbine failures but they are beyond the scope of this work.

In reduction gears, the driving elements are tooth bending stress and maximum compressive stress, so these stresses will be studied to determine what effects changes in reduction gear size, power, and reduction ratio have on MTBF.

Blade stresses in waterjet pumps and propellers will also be studied to determine the effects of size, RPM, and powering changes. This chapter is intended to determine the trends these changes produce in stress levels and MTBF and not the specific numerical solutions. Design standards and materials are assumed to remain the same.

4.2 Gas Turbine MTBF

Four of the five ships addressed in this work have LM2500 gas turbines with a 22,000 SHP max continuous rating. One of these ships only operates the gas turbine at 16,000 SHP maximum. It is expected that operating at such a reduced load, the gas turbine will have a longer MTBF. The effects of loading and changes in gas turbine size will be addressed in the following sections.

4.2.1 Power Level Affects on Gas Turbine MTBF

The stresses in a gas turbine blade can be divided into three categories as follows:

--centrifugal stress

--gas bending stress

--vibrational stress

Centrifugal stress σ_{CT} is defined by the following equation:

$$\sigma_{CT} = 2\pi N^2 A \rho_b$$

where

N = rotational speed in RPS

A = annular area

ρ_b = blade material density

Likewise, gas bending stress σ_{GB} is approximated by:

$$\sigma_{GB} \approx \frac{\dot{m}\, C_a (\tan \alpha_2 + \tan \alpha_3) H}{2n(Zc^3)}$$

where

n = number of blades

Zc^3 = blade section modulus

C_a = gas axial velocity

α_2 = angle between absolute velocity at rotor inlet and axial direction

α_3 = swirl angle

H = blade height

$\dot{m}$ = mass flow rate

The stage work per unit flow W_S

$$W_S = U\ C_a(\tan\alpha_2 + \tan\alpha_3)$$

where

U = blade speed at radius R = NR

$$W_S = \frac{\Delta h_{STAGE}}{\dot{m}} = \frac{SHP}{n\ \dot{m}}$$

where

SHP = total turbine power output

$$C_a(\tan\alpha_2 + \tan\alpha_3) = \frac{SHP}{U\ n\ \dot{m}}$$

Therefore σ_{GB} can be rewritten as

$$\sigma_{GB} = \frac{SHP}{N} \times \frac{H}{2\ n^2\ R(Zc^2)}$$

Vibrational stress σ_v is calculated by

$$\sigma_v = 1.3\ M_f\ \sigma_{GB}$$

where

M_f = multiplication factor = $\frac{41xN}{f_n}$

f_n = blade natural frequency

$$\sigma_v = \frac{53.3}{f_n}\ N\ \sigma_{GB}$$

$$\sigma_v = SHP \times \frac{26.65\ H}{n^2\ R\ f_n (Zc^3)}$$

For a given gas turbine design such as the LM2500, the following variables are fixed by the design:

A	ρ_b	f_n	R
(Zc^3)	H	n	

Also, for naval applications the power output of the LM2500 varies proportionately to the cube of RPM.

$$SHP\ \alpha\ RPM^3$$

With the above constants, and the relationship between power and RPM the following proportionalities can be derived:

$$\sigma_{CT} \propto SHP^{2/3}$$

$$\sigma_{GB} \propto SHP^{2/3}$$

$$\sigma_{V} \propto SHP$$

Therefore, as power requirements are reduced the total stress in the blade σ is reduced.

$$\sigma = \sigma_{V} + \sigma_{GB} + \sigma_{CT}$$

When the maximum power output of a given gas turbine is reduced due to some system constraint, the stress levels will also be reduced. This is not the only measure of MTBF however. Blade elongation is also a measure of failure so the creep rate as a function of power level must be determined to assess the overall effects.

Creep rate is a function of temperature, as temperature increases, creep rate does likewise. Therefore, how does turbine inlet temperature vary with power level? A typical gas turbine power cycle is illustrated in Figure 5.

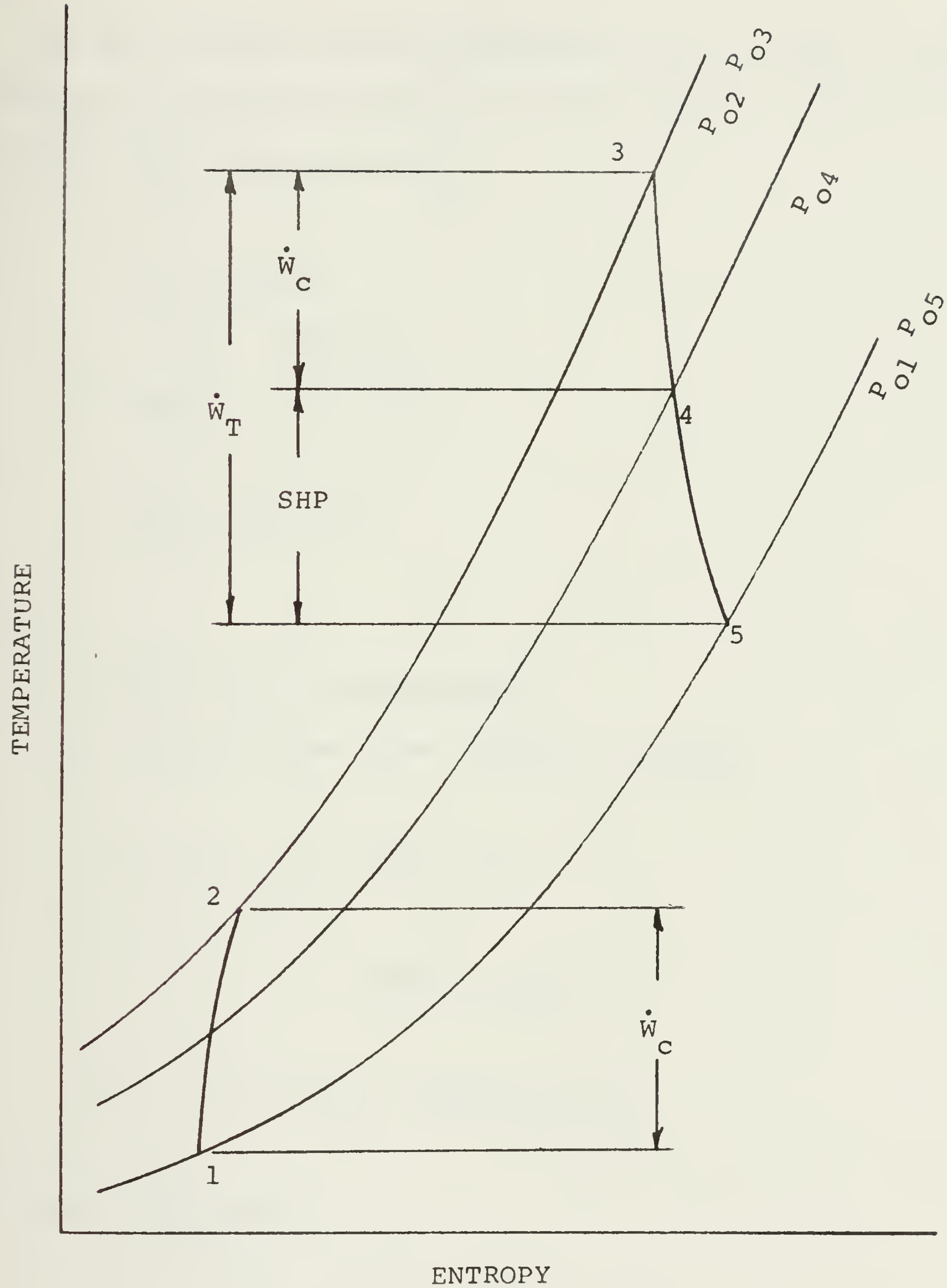

FIGURE 5 - TEMPERATURE-ENTROPY DIAGRAM FOR AN OPEN CYCLE GAS TURBINE

In this analysis some assumptions will be made. It is assumed that the mass flow rate, cycle pressure ratio, and gas properties remain constant.

$$\dot{m} = \text{constant}$$

$$\gamma,\ Cp = \text{constant}$$

$$P_{o2}/P_{o1} = \text{constant}$$

$$\dot{W}_T = SHP + \dot{W}_c$$

where

$\dot{W}_c$ = power to drive compressor

SHP = output power from turbine to system

$$\dot{W}_c = \dot{m}\ \Delta\ h_o = \dot{m}(h_{o2}-h_{o1})$$

$$= \dot{m}Cp(T_{o2}-T_{o1})$$

$$\dot{W}_T = \dot{m}(h_{o3}-h_{o5}) = \dot{m}Cp(T_{o3}-T_{o5})$$

For a compressor

$$\frac{T_{o2}}{T_{o1}} = [\frac{P_{o2}}{P_{o1}}]^{\gamma-1/\eta_{pc}\ \gamma}$$

Since P_{o2}/P_{o1}, γ, and η_{pc} are constant

$$\frac{T_{o2}}{T_{o1}} = \text{constant} = K_1$$

$$\dot{W}_c = \dot{m}Cp(K_1-1)T_{o1}$$

For a turbine

$$\frac{T_{o5}}{T_{o3}} = \left[\frac{P_{o5}}{P_{o3}}\right]^{\eta_{PT}\frac{\gamma-1}{\gamma}}$$

$$\frac{P_{o5}}{P_{o3}} = \frac{P_{o2}}{P_{o1}} = \text{constant}$$

$$\gamma,\ \eta_{PT} = \text{constants}$$

$$\frac{T_{o5}}{T_{o3}} = \text{constant} = K_2$$

$$\dot{W}_T = \dot{m}Cp(1-K_2)T_{o3}$$

$$\dot{m}Cp(1-K_2)T_{o3} = SHP + \dot{m}Cp(K_1-1)T_{o1}$$

$$T_{o3} = \frac{SHP + \dot{m}Cp(K_1-1)T_{o1}}{\dot{m}Cp(1-K_2)}$$

T_{o1} is the compressor inlet temperature which is ambient atmospheric and will be the same no matter what the power level. Thus,

$$T_{o3} = \frac{SHP + a}{b}$$

This result demonstrates that as the gas turbines output power is reduced the turbine inlet temperature is reduced as well.

Creep deformation with a constant stress is possible at all temperatures above absolute zero. However since it depends on thermal activation, the strain rate at a given stress level is extremely temperature sensitive. As a result, the higher the temperature, the more significant becomes the creep phenomena.

The creep rate $\dot{E}$ is defined by

$$\dot{E} = A\ e^{-q/KT}$$

where

A = constant

q = constant

K = property of material

T = temperature

For a given gas turbine design, if maximum power is reduced, stress is reduced, turbine inlet temperature is reduced and therefore the creep rate is reduced. Thus, the gas turbine operates at lower stresses and will expect to have a longer life from a reduced rate of creep elongation.

4.2.2 Gas Turbine Size Affects on MTBF

This section investigates the effects an increase in gas turbine size has on MTBF. The following assumptions are made:

SHP = constant

$\dot{m}$ = constant

velocity diagrams remain geometrically similar

Since the size is increasing, the RPM should decrease.

Specific speed N_s is defined by

$$N_s = \frac{N}{60} \frac{\sqrt{Q}}{(g_o \; J \; \Delta h_o)^{3/4}}$$

$$= K_3 \frac{N\sqrt{Q}}{(\Delta h_o)^{3/4}}$$

where

Q = volume flow rate

N = RPM

Δh_o = work done

$$Q = C_a A = \frac{\dot{m}}{\rho} = \text{constant}$$

$$\Delta h_o = \frac{SHP}{\rho C_a A}$$

$$N_s = K_3 N \frac{(C_a A)^{5/4}}{(SHP/\rho)^{3/4}}$$

The flow coefficient ϕ

$$\phi = \frac{C_a}{U} = \frac{C_a}{\pi ND} = \text{constant} = K_4$$

$$A \approx \pi D H_b$$

$$\dot{m} = \rho \pi^2 N D^2 H_b = \text{constant}$$

where

H_b = blade height

N_s shall be kept equal to the value used if the turbine size were smaller.

$$N_s = \text{constant} = K_5$$

$$K_5 = N^{9/4} \frac{K_6}{(SHP/\rho)^{3/4}} (D^2 H_b)^{5/4}$$

$$N = [\frac{(SHP/\rho)^{3/4}}{K_7} \frac{1}{(D^2 H_b)^{5/4}}]^{4/9}$$

or

$$N \alpha (\frac{1}{D^2 H_b})^{5/9}$$

With the assumption that H_b will increase with increasing turbine diameter:

$$N \alpha (\frac{1}{D^3})^{5/9}$$

$$N \alpha (\frac{1}{D})^{5/3}$$

This relationship shows that the gas turbine RPM decreases with increasing size.

Using the same equations for stress that were given in the previous section, the resulting effects on stress are evaluated.

Centrifugal stress σ_{CT}

$$\sigma_{CT} \propto D^2 N^2$$

$$\sigma_{CT} \propto D^{-4/3}$$

Gas bending stress σ_{GB}

$$\sigma_{GB} \approx \frac{SHP}{N} \times \frac{H}{n^2 D(Zc^3)}$$

When the diameter increases the number of blades increase, as related by:

$$n \propto \pi D$$

$$\sigma_{GB} \propto D^{-11/3}$$

Vibrational stress σ_v

$$\sigma_v = 1.3\left(\frac{41 x N}{f_n}\right) \sigma_{GB}$$

$$f_n \propto \frac{1}{H^2} \propto \frac{1}{D^2}$$

$$\sigma_v \propto D^{-10/3}$$

Therefore, as the size of a gas turbine increases for constant power output, RPM decreases and total stress decreases. The net result is then a longer expected MTBF.

4.3 Reduction Gear MTBF

This section will investigate changes in MTBF brought about by varying component size and reduction gear ratio independently, and then in combination. The reduction gears are assumed to be constructed of the same material and no special design or manufacturing technique are employed.

4.3.1 Size Affects on Reduction Gear MTBF

In reduction gears, tooth contact pressure determines the durability of the working surfaces of the teeth.

$$\frac{W_t}{F_e} = 126{,}050 \frac{SHP}{RPM_p \times d \times F_e}$$

where

W_t = total tangential tooth load

F_e = effective face width at pitch diameter

RPM_p = pinion revolutions per minute

SHP = horsepower transmitted per mesh

d = pitch diameter of pinion

It should be noted that allowable tooth load increases as pinion pitch diameter increases due to the decreasing curvature of the contacting surfaces.

$$\frac{W_t}{F_e}\text{(allowable)} = K\frac{R}{R+1}\,d$$

where

R = gear ratio

K = experimentally determined constant

$$K = \frac{126{,}500 \times SHP}{RPM_p \times d^2 \times F_e} \times \frac{R+1}{R}$$

It so happens that K is a good measure of tooth surface stress (maximum compressive stress) to which the tooth materials in contact are subjected.

Tooth bending stress is evaluated by the expression

$$S_b = k\,\frac{W'_n}{Y}$$

where

W'_n = loading per inch of contact line

y = tooth form factor = $t^2/6h$

h = tooth height

t = tooth thickness at root

k = diagonal loading factor which varies with helix angle

$$W'_n = \frac{W_t \, P_n}{F_e \, Z \cos\phi_n \cos\psi}$$

$$= 126{,}050 \times \frac{SHP \times P_n}{RPM_p \times d \times F_e \times Z \cos\phi_n \cos\psi}$$

where

ϕ_n = pressure angle

ψ = helix angle

P_n = normal base pitch

Z = length of line of action

Therefore, tooth bending stress S_b

$$S_b = 756{,}300 \, k \, \frac{SHP}{RPM_p} \, \frac{P_n \, h}{F_e \times d \times t^2 \times Z \cos\phi_n \cos\psi}$$

In order to determine the effects of decreasing the reduction gear size, a few assumptions must be made. The tooth geometry, power level, and RPM remain fixed. Also, to avoid excessive pinion deflections, $F_e/d = 2.25$.

Therefore, the compressive stress and tooth bending stress equations can be written as:

$$K \propto \frac{1}{d^3}$$

$$S_b \propto \frac{1}{d^2}$$

These expressions show that as the reduction gear size decreases, both maximum compressive stress and tooth bending stress increase. This increase in stress will cause MTBF to be reduced.

4.3.2 Reduction Ratio Affects on MTBF

The influence reduction gear ratio variation has on component MTBF is investigated in this section.

Reduction ratio is defined by

$$R = \frac{D}{d} = \frac{RPM_B}{RPM_p}$$

where

D = bull gear diameter

d = pinion diameter

RPM_B = bull gear RPM

RPM_p = pinion RPM

If all variables remain constant as assumed in the previous section, when reduction ratio decreases either the pinion diameter must increase or the bull gear diameter must decrease. Both cases are of interest.

Holding the bull gear diameter constant and increasing the pinion diameter, the reduction ratio increases.

Since $\frac{R+1}{R} = \frac{D+d}{D}$

as d increases the ratio $\frac{R+1}{R}$ also increases.

The effects on maximum compressive and tooth bending stresses are demonstrated by the following relationships:

$$K \propto \frac{1}{d^2} \times \frac{R+1}{R} = \frac{1}{d^2} \times \frac{D+d}{D}$$

$$\propto \frac{a}{d^2} + \frac{b}{d}$$

$$S_b \propto \frac{1}{d^2}$$

Therefore, decreasing reduction ratio by increasing pinion diameter will result in decreasing both maximum compressive and tooth bending stresses.

Holding the pinion diameter constant, reduction ratio is decreased by decreasing bull gear diameter.

$\frac{R+1}{R}$ increases as R decreases.

The maximum compressive stress is now dependent only on R+1/R, and tooth bending stress independent of R.

$$K \propto \frac{R+1}{R}$$

$$S_b = \text{constant}$$

Thus, as reduction ratio is decreased by decreasing bull gear diameter, there is no change in tooth bending stress and an increase in maximum compressive stress.

4.3.3 Combined Size and Reduction Ratio Affects on MTBF

In assessing the changes expected in MTBF when going from a large double reduction gear train to a small single reduction gear box, as is the case when going from conventional displacement to high performance propulsion designs,

the combination of size reduction and reduction ratio increase must be addressed.

Here, both bull gear and pinion diameters are decreasing simultaneously while their individual rates of decrease result in a decrease in reduction ratio as well.

All the previous derivations have shown that maximum compressive stress is related to pinion diameter and reduction ratio, while the tooth bending stress is dependent on pinion diameter alone.

Thus,

$$K \propto \frac{1}{d^2} \times \frac{R+1}{R}$$

$$S_b \propto \frac{1}{d^2}$$

With both pinion diameter and reduction ratio decreasing, the maximum compressive stress and tooth bending stress are increasing. Therefore, it is reasonable to expect the smaller component to have higher stress levels and as a result, a shorter MTBF than an equally rated large component.

4.4 Propulsor MTBF

The use of different propulsor types does not make for a convenient analysis of relative stress levels. However, the effects of varying size and power level will be investigated for both waterjet pumps and propellers individually.

4.4.1 Propellers

As done previously with gas turbines and reduction gears, size and power level effects on MTBF will be investigated for propellers.

4.4.1.1 - Size Affects on MTBF

Stresses in propeller blades are divided into three categories:

--maximum tensile stress

--maximum compressive stress

--centrifugal stress

Maximum tensil stress is defined by the relationship:

$$\sigma_T = \left[\frac{g}{k_c} \times \frac{C}{\ell t^2} + \frac{1}{2k_\ell} \times \frac{L}{\ell^2 t}\right] \frac{P_1}{N}$$

Where

g, k_c, and k_ℓ are coefficients dependent on the type of blade section

ℓ= length of blade section

t = blade thickness

L,C = coefficients dependent on pitch ratio and distance of section from hub centerline

P_1 = horsepower absorbed by a single blade

N = revolutions per minute

Maximum compressive stress is determined by

$$\sigma_c = \frac{(1-g)}{k_c \ \ell t^2} \times C \times \frac{P_1}{N}$$

Centrifugal stress is

$$\sigma_{CT} = K \rho d^2 N^2$$

where

K = coefficient dependent on distance from hub centerline

ρ = blade material density

d = blade diameter

When the propeller size is increased, it is assumed that SHP and RPM remain fixed and that the new blade is geometrically similar to the smaller one.

With these assumptions, maximum tensile and compressive stresses, and centrifugal stress equations reduce to:

$$\sigma_T \propto \frac{1}{d^3}$$

$$\sigma_C \propto \frac{1}{d^3}$$

$$\sigma_{CT} \propto d^2$$

As the propeller size increases, maximum tensile and compressive stress will decrease but centrifugal stress increases. The net result is not obvious with this type of analysis. For low RPM propellers, centrifugal stress is at least an order of magnitude less than maximum tensil or compressive stress so that a decrease in total stress is observed as diameter increases. Once again, increased size and weight of a component results in lower stresses and longer MTBF.

4.4.1.2 - Power Level Affects on MTBF

If propeller size is fixed, but it is operated at a fraction of its rated capacity, the stress equations show:

$$\sigma_T \; \alpha \; SHP$$

$$\sigma_C \; \alpha \; SHP$$

$$\sigma_{CT} \; \alpha \; \text{constant}$$

Thus, a propeller continually operating at a fraction of its designed rating should have lower stress levels and a longer MTBF than the same propeller operating at its maximum capacity. This change in stress level is not as dramatic as that produced by a change in propeller diameter.

4.4.2 Waterjet Pumps

The waterjet pumps in use on high performance ships such as the PHM-1 and LSES are axial flow and ressemble an axial flow compressor. This fact makes the analysis of blade stress identical to that of the gas turbine blades with the exception that in this case the blade is doing work on the fluid and the fluid density is much higher.

4.4.2.1 - Power Level Affects on MTBF

The equations for centrifugal, bending, and vibrational stress still hold from the gas turbine analysis and are summarized here.

$$\sigma_{CT} = 2\pi\ N^2\ A\rho_b$$

$$\sigma_B = \frac{SHP}{N} \times \frac{H}{2n^2\ R(Zc^2)}$$

$$\sigma_v = SHP \times \frac{26.65H}{n^2\ Rf_n(Zc^3)}$$

Fixing the same variables as before and noting SHP α RPM^3 leads to the proportionalities:

$$\sigma_{CT}\ \alpha\ SHP^{2/3}$$

$$\sigma_B\ \alpha\ SHP^{2/3}$$

$$\sigma_v\ \alpha\ SHP$$

The stresses decrease when the power level is reduced. Thus a waterjet pump operating continually below its rated power can be expected to exhibit a longer MTBF than the same propulsor operating at its maximum rated capacity.

4.4.2.2 - Size Affects on MTBF

The same relationship for specific speed indicates that as diameter increases, the working RPM will decrease.

The stresses are now dependent on diameter and RPM while power level and flow rate are the same. Geometrically similar velocity triangles are maintained as diameter increases.

Therefore,

$$\sigma_{CT} \propto D^{-4/3}$$

$$\sigma_{B} \propto D^{-11/3}$$

$$\sigma_{V} \propto D^{-10/3}$$

The stresses decrease as diameter increases, which results in the expected MTBF of the large component being greater than that of a similarly rated small component.

4.5 Summary and Conclusions

In every case considered, it has been demonstrated that decreasing the size of a component increases the stress levels encountered. This could lead to the conclusion that high performance propulsion systems are less reliable. However, in many cases high performance propulsion components have been designed and manufactured using the most advanced techniques and higher strength materials than used in conventional designs. This can counter the stress increases developed by size reduction. The higher strength materials and advanced manufacturing techniques allow stress levels to be increased without having an adverse effect on MTBF. The advanced design techniques are more precise and allow much of the conservatism to be eliminated.

In the cases where the same component is operated in different systems under different ratings, it has been demonstrated that variations in MTBF should be expected. Therefore, system designers should evaluate the effects on MTBF due to component rating and vary the MTBF appropriately when predicting system reliability and availability.

CHAPTER 5

MEAN-TIME-TO-REPAIR

The mean-time-to-repair used when predicting propulsion system availability accounts only for the elapsed time in performing maintenance in a prescribed manner. The immediate availability of spare parts, tools, and personnel, and good machinery accessability are assumed. In actual operation, this MTTR is seldom realized. The amount of spare parts and tools, the number of maintenance personnel, the accessability to equipment, and shop capability are all limited by constraints on weight and volume. These factors contribute significantly to the MTTR.

This chapter will compare the differences in ship characteristics which demonstrate the effects on MTTR and determine if maintainability in the form of MTTR is degraded on high performance ships. Indices will be developed to assess the effects of crew size and utilization, storage volume, shop capabilities, and machinery accessability.

5.1 MTTR Parameters

This section will present the parameters useful in evaluating differences in ship and propulsion system designs which directly influence the component and system MTTR's.

5.1.1 Manpower Parameters

Table 4 provides a listing of parameters used in evaluating difference in manpower which effect MTTR. The time fractions are an indication of the designer's assessment of various maintenance and support requirements.

Specific ratios such as HRS/SHP and HRS/HRS_{UW} indicate the "cost" in man hours of the specific SHP installed and the hours underway per week. The degree of utilization of each man is measured by HRS/M_E. A measure of the cost in crew hours for one hour of underway time normalized by propulsion plant capacity is given by HRS/SHP HRS_{UW}.

The difference between these parameters for high performance and conventional displacement ships will be investigated in the following sections.

5.1.2 Spare Parts Parameters

A listing of parameters useful in evaluating differences in spare parts availability are listed in Table 5. The priority given to spare parts and special tools are illustrated by the weight and volume fractions. The specific ratios of W_{299}/SHP and V_{STRM}/SHP indicate the amount of propulsion system support in the form of spare parts and tools normalized by system capacity.

TABLE 4

MANPOWER PARAMETERS

$\frac{HRS_{CM}}{HRS}$	Corrective maintenance time fraction
$\frac{HRS_{PM}}{HRS}$	Preventive maintenance time fraction
$\frac{HRS_{FM}}{HRS}$	Facility maintenance time fraction
$\frac{HRS_{WS}}{HRS}$	Watchstanding time fraction
$\frac{HRS}{SHP}$	Total propulsion division manhours normalized by propulsion system capacity
$\frac{HRS_{PM}}{SHP}$	Preventive maintenance manhours normalized by propulsion system capacity
$\frac{HRS_{CM}}{SHP}$	Corrective maintenance manhours normalized by propulsion system capacity
$\frac{HRS_{FM}}{SHP}$	Facility maintenance manhours normalized by propulsion system capacity
$\frac{HRS_{WS}}{SHP}$	Watchstanding manhours normalized by propulsion system capacity

TABLE 4 (cont)

$\frac{HRS}{M_E}$	Weekly hours per man
$\frac{HRS}{M_E\ HRS_{UW}}$	Hours per man per hour underway
$\frac{HRS}{SHP\ HRS_{UW}}$	Total propulsion division manhours normalized by propulsion system capacity and hours of use
$\frac{HRS_{PM}}{SHP\ HRS_{UW}}$	Preventive maintenance hours normalized by propulsion system capacity and hours of use
$\frac{HRS_{CM}}{SHP\ HRS_{UW}}$	Corrective maintenance hours normalized by propulsion system capacity and hours of use
$\frac{HRS_{FM}}{SHP\ HRS_{UW}}$	Facility maintenance hours normalized by propulsion system capacity and hours of use
$\frac{HRS_{WS}}{SHP\ HRS_{UW}}$	Watchstanding hours normalized by propulsion system capacity and hours of use
$\frac{HRS_{PM}}{HRS_{CM}}$	Ratio of preventive maintenance to corrective maintenance

TABLE 5

SPARE PARTS PARAMETERS

$\frac{W_{299}}{W_2}$	Propulsion spare part/special tools weight fraction
$\frac{W_{299}}{\Delta}$	Spare part/special tools weight fraction
$\frac{W_{299}}{SHP}$	Propulsion spare parts/special tools weight normalized by system capacity
$\frac{V_{STRM}}{V_2}$	Storeroom-propulsion volume fraction
$\frac{V_{STRM}}{\nabla}$	Storeroom volume fraction
$\frac{W_{299}}{V_{STRM}}$	Spare part-special tool density
$\frac{V_{STRM}}{SHP}$	Spare parts/special tools volume normalized by system capacity

5.1.3 Machinery Access Parameters

Parameters measuring the compactness of a machinery space are presented in Table 6. Deck area and volume fractions indicate the relative amount of accessability each system possesses.

5.1.4 Repair Shop Capability Parameters

Table 7 presents the applicable parameters measuring repair shop capability. The propulsion system related shop volume fraction indicates the priority given to repair shops in the ship design. The other parameters evaluate shop volume and manpower normalized by propulsion system capacity or weight.

5.2 Propulsion System MTTR Assessment

Using the parameters developed in the previous section, the influence these factors have on MTTR can be evaluated for high performance and conventional displacement ships. Comparing the differences will indicate the relative maintainability of each type of propulsion system.

5.2.1 Manpower Affects on MTTR

Table 8 summarizes propulsion division manhours from each ship's respective manning document. The number of men in the division and days normally underway each week

TABLE 6

MACHINERY ACCESSABILITY PARAMETERS

$\frac{A_{MACH}}{A_{M.S.}}$	Machinery deck area allocation
$\frac{V_{MACH}}{V_{M.S.}}$	Machinery volume allocation in space

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TABLE 7

REPAIR SHOP CAPABILITY PARAMETERS

$\frac{V_{E.S.}}{SHP}$	Propulsion related repair shop volume normalized to system capacity
$\frac{M_E}{SHP}$	Propulsion division manning normalized to system capacity
$\frac{M_E}{SHP}$	Repair division (propulsion related) manning normalized to system capacity
$\frac{M_E}{W_2}$	Propulsion division manning normalized to system weight
$\frac{M_R}{W_2}$	Repair division (propulsion related) manning normalized to system weight
$\frac{V_{E.S.}}{\nabla}$	Propulsion related repair shop volume fraction

TABLE 8

SUMMARY OF PROPULSION DIVISION MANHOURS FOR ONE WEEK

SHIP	DAYS U.W. PER WK.	PROP. DIV. CREW	PM	MR/PA	CM	FM	COND III WATCH STAND.	OTHER	ALLOW.	TOTAL WEEKLY HOURS
PHM-1	5	6	92.77	27.83	46.38	7.00	360.00	8.00	20.3	562.28
PG-84	5	12	199.10	59.73	76.60	22.00	336.00	33.00	127.14	853.57
LSES	7	6	19.96	5.98	36.96	18.00	168.00	45.24	79.42	373.56
FFG-7	7	10	56.90	17.07	28.42	45.26	392.00	73.10	95.15	707.9
DD-963	7	27	217.42	83.16	59.75	151.21	616.00	95.55	256.95	1480.04

PM - preventative maintenance

CM - corrective maintenance

FM - facility maintenance

MR/PA - make ready/put away, 30% of PM

COND III WATCH STANDING - watch standing

OTHER - admin support and utility tasks

ALLOWANCES - 20% production allowance, training and service diversion

are also indicated. The large combatants are underway for several weeks at a time, so they show all seven days of the week underway. Normally, the small ships operate only five days out of each week.

Manpower indices are evaluated from this data and compared in Table 9. The similarity between PHM-1 and PG-84 propulsion systems, as well as those of LSES and DD-963 should be remembered as the manpower differences are evaluated.

5.2.1.1 - Small Ships

When evaluating the allocation of crew time, the observation is made that small, high performance and conventional ships give equal priority to corrective maintenance.

	PHM	PG-84
Corrective Maintenance Time Fraction	8.25%	8.97%
Preventive Maintenance Time Fraction	16.50%	23.31%
Facility Maintenance Time Fraction	1.24%	2.58%
Watchstanding Time Fraction	64.03%	39.36%

However, much smaller percentages of crew time are available for preventive and facility maintenance. Since the propulsions systems are quite similar, either the

TABLE 9

MANPOWER INDICES

PARAMETER	UNITS	PHM-1	PG-84	LSES	FFG-7	DD-963
$\frac{HRS_{CM}}{HRS}$	%	8.25	8.97	9.89	4.01	4.04
$\frac{HRS_{PM}}{HRS}$	%	16.50	23.31	5.34	8.04	14.69
$\frac{HRS_{FM}}{HRS}$	%	1.24	2.58	4.80	6.39	10.22
$\frac{HRS_{WS}}{HRS}$	%	64.03	39.36	44.97	55.38	41.62
$\frac{HRS_{PM}}{HRS_{CM}}$	-	2.00	2.60	0.54	2.00	3.64
$\frac{HRS}{SHP}$	$\frac{HRS/WK}{10^3\ SHP}$	25.42	55.07	4.34	17.27	17.21
$\frac{HRS_{PM}}{SHP}$	"	4.19	12.85	0.23	1.39	2.53
$\frac{HRS_{CM}}{SHP}$	"	2.10	4.94	0.43	0.69	0.69
$\frac{HRS_{FM}}{SHP}$	"	0.32	1.42	0.21	1.10	1.76
$\frac{HRS_{WS}}{SHP}$	"	16.27	21.68	1.95	9.56	7.16
$\frac{HRS}{M_E}$	$\frac{HRS/WK}{MAN}$	93.71	71.13	62.26	70.79	54.82
$\frac{HRS}{M_E\ HRS_{UW}}$	$\frac{HRS/MAN}{HRS_{UW}}$	0.78	0.59	0.37	0.42	0.33

TABLE 9 (cont)

Parameter	Units	PHM-1	PG-84	LSES	FFG-7	DD-963
$\frac{HRS}{SHP\ HRS_{UW}}$	$\frac{HRS}{10^3 SHP\ HRS_{UW}}$	0.21	0.46	0.026	0.103	0.102
$\frac{HRS_{PM}}{SHP\ HRS_{UW}}$	"	0.35	1.07	0.014	0.083	0.15
$\frac{HRS_{CM}}{SHP\ HRS_{UW}}$	"	0.18	0.41	0.026	0.041	0.042
$\frac{HRS_{FM}}{SHP\ HRS_{UW}}$	"	0.03	0.12	0.013	0.066	0.105
$\frac{HRS_{WS}}{SHP\ HRS_{UW}}$	"	1.36	1.81	0.116	0.57	0.43

difference in time allocation is due to lower preventive maintenance requirements of the reduction gear and waterjet pump or the crew's time is demanded by another function. The latter is the case, since it is observed that the crew of the PHM devotes over 60% of their time to watch standing. This is necessary since the PHM has half as many watchstanding personnel as the PG-84.

The time allocated per function as normalized by propulsion system capacity indicates the cost in crew time for each SHP of plant capacity.

	PHM		PG-84	
$\frac{HRS}{SHP}$	25.42	$\frac{HRS/WK}{SHP \times 10^3}$	55.07	$\frac{HRS/WK}{SHP \times 10^3}$
$\frac{HRS_{PM}}{SHP}$	4.19	"	12.85	"
$\frac{HRS_{CM}}{SHP}$	2.10	"	4.94	"
$\frac{HRS_{FM}}{SHP}$	0.32	"	1.42	"
$\frac{HRS_{WS}}{SHP}$	16.27	"	21.68	"

The crew of the conventional ship is able to devote more time to each function relative to plant size than is the crew of the high performance ship. The conventional displacement crew can perform three times the preventive maintenance which should enhance the MTBF of the system. They are able to accomplish 2.5 times more corrective maintenance and four times the facility maintenance, as related to plant capacity.

The hours per week per man and the hours per man per underway hour give a good indication of the utilization of manpower and the fraction of each underway hour a man must devote to propulsion system operation and maintenance.

	PHM	PG-84
$\frac{HRS}{M_E}$	93.71 $\frac{HRS/WK}{MAN}$	71.13 $\frac{HRS/WK}{MAN}$
$\frac{HRS}{M_E\ HRS_{UW}}$	0.78 $\frac{HRS/MAN}{HRS_{UW}}$	0.59 $\frac{HRS/WK}{HRS_{UW}}$

As shown by the ratio HRS/M_E, each member of the PHM's propulsion division must work over 20 hours more per week than a man on the PG-84. This is about 13.5 hours a day per man. Any additional demand on the crew's time should have adverse affects on their effectiveness. This high usage

of manpower for normal operation is evident in the ratio

$$\frac{HRS}{M_E\ HRS_{UW}}.$$

The manhours normalized by plant capacity and hours per week underway illustrate the significantly degraded maintenance capability of the PHM propulsion division as compared to the PG-84.

	PHM	PG-84
$\frac{HRS}{SHP\ HRS_{UW}}$	0.21 $\frac{HRS}{10^3 SHP\ HRS_{UW}}$	0.46 $\frac{HRS}{10^3 SHP\ HRS_{UW}}$
$\frac{HRS_{PM}}{SHP\ HRS_{UW}}$	0.35 "	1.07 "
$\frac{HRS_{CM}}{SHP\ HRS_{UW}}$	0.18 "	0.41 "
$\frac{HRS_{FM}}{SHP\ HRS_{UW}}$	0.03 "	0.12 "
$\frac{HRS_{WS}}{SHP\ HRS_{UW}}$	1.36 "	1.81 "

The ratio of preventive maintenance hours to corrective maintenance hours indicates that the conventional displacement ship does more preventive maintenance relative to

corrective maintenance. This is consistent with the time fractions discussed previously.

5.2.1.2 - Large Ships

In evaluating time fractions for the large ships, it is observed that in only a few cases are the priorities of types of maintenance functions similar for two or more ships. The FFG-7 and DD-963 give equivalent priority to corrective maintenance but the LSES gives it a much higher priority. LSES and DD-963 devote nearly the same fraction of crew time to watchstanding. Other than these few similarities, each design assesses the importance of these maintenance and operational areas differently.

	LSES	FFG-7	DD-963
Corrective Maintenance Time Fraction	9.89%	4.01%	4.04%
Preventive Maintenance Time Fraction	5.34%	8.04%	14.69%
Facility Maintenance Time Fraction	4.80%	6.39%	10.22%
Watchstanding Time Fraction	44.97%	55.38%	41.62%

The time allocated for each function as normalized by the propulsion plant capacity indicates the time the crew must devote for each SHP of plant capacity.

	LSES		FFG-7		DD-963	
$\frac{HRS}{SHP}$	4.34	$\frac{HRS/WK}{10^3\ SHP}$	17.27	$\frac{HRS/WK}{10^3\ SHP}$	17.21	$\frac{HRS/WK}{10^3\ SHP}$
$\frac{HRS_{PM}}{SHP}$	0.23	"	1.39	"	2.53	"
$\frac{HRS_{CM}}{SHP}$	0.43	"	0.69	"	0.69	"
$\frac{HRS_{FM}}{SHP}$	0.21	"	1.10	"	1.76	"
$\frac{HRS_{WS}}{SHP}$	1.95	"	9.56	"	7.16	"

In every case, the conventional displacement ship crew is able to devote substantially more time to maintenance and watchstanding as related to propulsion system capacity. With the time devoted to the conventional system being four times that of an equally rated high performance propulsion system, the conventional ships should have better operability.

The work week per man varies for all three ships. The work week for the LSES crew is about the average of that for the FFG-7 and DD-963.

	LSES		FFG-7		DD-963	
$\frac{HRS}{M_E}$	62.26	$\frac{HRS/WK}{MAN}$	70.79	$\frac{HRS/WK}{MAN}$	54.82	$\frac{HRS/WK}{MAN}$
$\frac{HRS}{M_E\ HRS_{UW}}$	0.37	$\frac{HRS/MAN}{HRS_{UW}}$	0.42	$\frac{HRS/MAN}{HRS_{UW}}$	0.33	$\frac{HRS/MAN}{HRS_{UW}}$

The same is true for the hours per man for each underway hour. The utility of each man is about equal for high performance and conventional ships; however, as shown by the ratio HRS/SHP, the conventional displacement ship crew is capable of four times the work. This is partially due to the larger crews aboard conventional displacement ships.

The ratios of HRS/SHP HRS_{UW} show that conventional displacement ships have more time devoted to their propulsion plants relative to capacity and hours of use than do high performance ships.

	LSES	FFG-7	DD-963
$\frac{HRS}{SHP\ HRS_{UW}}$	$0.26 \frac{HRS}{10^3 SHP\ HRS_{UW}}$	$0.103 \frac{HRS}{10^3 SHP\ HRS_{UW}}$	$0.102 \frac{HRS}{10^3 SHP\ HRS_{UW}}$
$\frac{HRS_{PM}}{SHP\ HRS_{UW}}$	0.014 "	0.083 "	0.15 "
$\frac{HRS_{CM}}{SHP\ HRS_{UW}}$	0.026 "	0.041 "	0.042 "
$\frac{HRS_{FM}}{SHP\ HRS_{UW}}$	0.013 "	0.066 "	0.105 "
$\frac{HRS_{WS}}{SHP\ HRS_{UW}}$	0.116 "	0.57 "	0.43 "

The low times devoted to these functions indicates the potential for poorer operability of the high performance propulsion system.

For both small and large ships, the high performance propulsion systems are manned by smaller crews who devote much less total time to operation and maintenance functions. Thus, the high performance propulsion systems should have longer MTTR's than their conventional counterparts.

5.2.2 Spare Parts Affect on MTTR

Indices defined for spare parts availability have been calculated for the high performance and conventional displacement ships and are listed in Table 10. It is noted that conventional displacement ships allot a larger percentage of their total volume to spare parts storage, thus they have the ability to carry more spares. Relative to propulsion system capacity, the conventional displacement ships have better spare parts availability as well. Therefore, high performance ships have poorer spare parts availability than conventional displacement ships which will increase MTTR and degrade system operability.

TABLE 10

SPARE PARTS INDICES

PARAMETERS	UNITS	PHM-1	PG-84	LSES	FFG-7	DD-963
$\frac{W_{299}}{W_2}$	%	1.28	0.43	0.23	0.71	1.14
$\frac{W_{299}}{\Delta}$	%	0.143	0.083	0.016	0.057	0.113
$\frac{V_{STRM}}{V_2}$	%	0.00	7.24	0.48	15.24	5.98
$\frac{V_{STRM}}{\nabla}$	%	0.00	1.81	0.07	2.10	1.17
$\frac{W_{299}}{V_{STRM}}$	$\frac{\text{POUNDS}}{\text{FT}^3}$	--	0.51	1.90	0.43	1.61
$\frac{W_{299}}{SHP}$	$\frac{\text{POUNDS}}{\text{x}10^3\ \text{SHP}}$	35.44	38.90	12.50	112.55	228.95
$\frac{V_{STRM}}{SHP}$	$\frac{\text{FT}^3}{\text{x}10^3\ \text{SHP}}$	0.00	56.9	6.59	263.41	142.53

TABLE 11

MACHINERY ACCESSABILITY INDICES

PARAMETERS	UNITS	PHM-1	PG-84	LSES	FFG-7	DD-963
$\frac{A_{MACH}}{A_{M.S.}}$	%	0.66	0.53	0.33	0.41	0.51
$\frac{V_{MACH}}{V_{M.S.}}$	%	0.43	0.25	0.26	0.30	0.43

5.2.3 Machinery Accessability Affects on MTTR

Hydrofoils, which have displacement hull forms, and conventional displacement ships are volume limited to some degree due to the hull form. Hydrofoils are weight limited with respect to their lift capability, however. The limit in volume has a constraining effect on access. Surface effect ships have excess volume by nature of their hull form and do not constrain access.

With these observations in mind, the difference in machinery accessability presented in Table 11 can be evaluated. The PHM is a very compact design leading to poorer accessability than the PG-84. The LSES, on the other hand, takes advantage of the available excess volume and actually has better accessability than the conventional ships.

Accessability can be enhanced in SES propulsion systems. The more compact hydrofoil propulsion system has reduced machinery accessability which causes the expected MTTR of components and systems to increase.

5.2.4 Repair Shop Capability Affects on MTTR

Differences in repair shop capability are evaluated using various indices in Table 12. It is observed that conventional ships allocate significantly more of their total volume to repair shops. This indicates the higher

TABLE 12

REPAIR SHOP CAPABILITY INDICES

PARAMETER	UNITS	PHM-1	PG-84	LSES	FFG-7	DD-963
$\frac{V_{E.S.}}{SHP}$	$\frac{FT^3}{x10^3\ SHP}$	0.00	13.94	27.52	147.51	140.23
$\frac{M_E}{SHP}$	$\frac{MEN}{x10^3\ SHP}$	0.27	0.77	0.07	0.24	0.31
$\frac{M_R}{SHP}$	$\frac{MEN}{x10^3\ SHP}$	0.00	0.065	0.023	0.122	0.058
$\frac{M_E}{W_2}$	$\frac{MEN}{x10^2\ TON}$	14.68	14.93	2.91	2.07	3.49
$\frac{M_R}{W_2}$	$\frac{MEN}{TON}$	0.00	0.21	0.97	1.72	0.65
$\frac{V_{E.S.}}{\nabla}$	%	0.00	0.44	0.30	1.17	1.15

priority given this function by conventional displacement ships. The shop capacity is also greater for conventional ships when related to the power output capacity of the propulsion system. Along with shop capacity, the conventional displacement ships allocate more men per ton of machinery for repair than do the high performance ships.

Overall, conventional ships have better manned, higher capacity repair shops than high performance ships. This superiority in repair facilities should reduce the conventional displacement ship propulsion system MTTR significantly.

5.3 Summary and Conclusions

From the indices calculated and presented in the previous sections, it has been shown that high performance vessels have many characteristics which have adverse effects on MTTR. It should be noted, however, that this analysis is limited to two high performance ships and the results may not always generally apply.

What should be concluded, however, is that there are many factors in a ship design which directly effect the maintainability of a component or system. A designer must not only design system configurations for high operability, but must also insure supporting areas of the design such as accessability, storage, and manpower do not degrade this operability.

The following chapter will investigate the differences in predicted R/M/A. It will be followed by a discussion concerning the designers' predictions of propulsion system R/M/A and the degree to which this prediction is supported in the design.

CHAPTER 6

R/M/A EVALUATION OF HIGH PERFORMANCE AND CONVENTIONAL PROPULSION SYSTEM CONFIGURATIONS

In the previous chapters, the factors which influence R/M/A have been discussed. Propulsion system configurations will now be evaluated to determine the degree of reliability and availability that are inherent in the high performance and conventional designs. The effects of redundancy will be studied first. Next, the differences in reliability and availability due to the type of propulsor will be investigated. Lastly, the reliability and availability predicted by the ship designers will be looked at and the differences between MTBF and MTTR data evaluated.

With this analysis completed, a comparison shall be made between predicted reliability and availability and what should actually be expected from the influences discussed in earlier chapters. Summing these effects, a determination is made as to whether reliability and availability are sacrificed to install low weight and volume propulsion systems.

6.1 Selection of Reliability and Availability Indices

The reliability and availability of a propulsion system are usually predicted for the full power configuration. This is very useful, but there is additional information to be gained by evaluating other power levels as well.

Figures 6 through 8 illustrate typical speed-power relationships for hydrofoil, surface effect, and conventional monohull ships. The high performance speed-power curves are characterized by a "hump". After the hump, the ship is in "flight" and gains a significant speed increase with only a small additional investment in power. This speed improvement is brought about by the reduced drag associated with the high performance ship lift system.

Figures 9 and 10 are speed-time distributions which indicate that high performance ships speed much of their operating time above half power in the reduced drag region. Thus, the best measure of operability for the high performance propulsion system is full power reliability.

Conventional displacement ships have speed-power curves illustrated by Figure 8. This speed-power relationship is characterized by a cubic relationship. As the conventional displacement ship increases speed, it must provide an ever increasing amount of power. Because speed is so costly in terms of power, the conventional displacement ship operates at power levels below half power, as substantiated by the speed-time distributions of Figures 11 and 12. The best measure of operability for the conventional displacement ship is half power reliability.

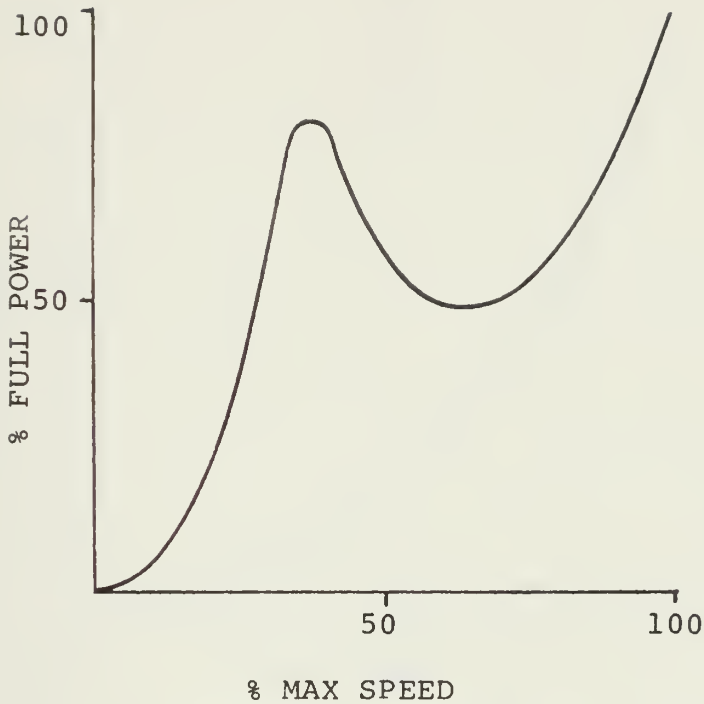


FIGURE 6 - TYPICAL HYDROFOIL POWERING CURVE

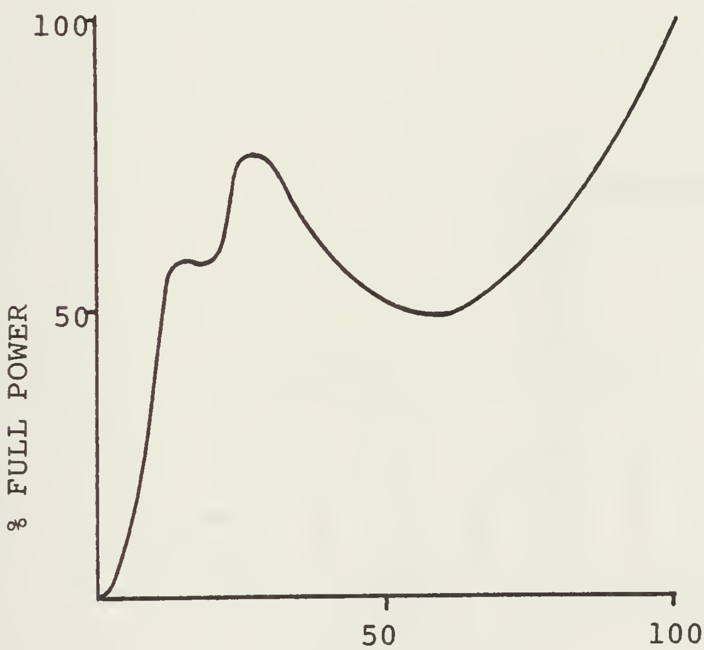


FIGURE 7 - TYPICAL SES POWERING CURVE

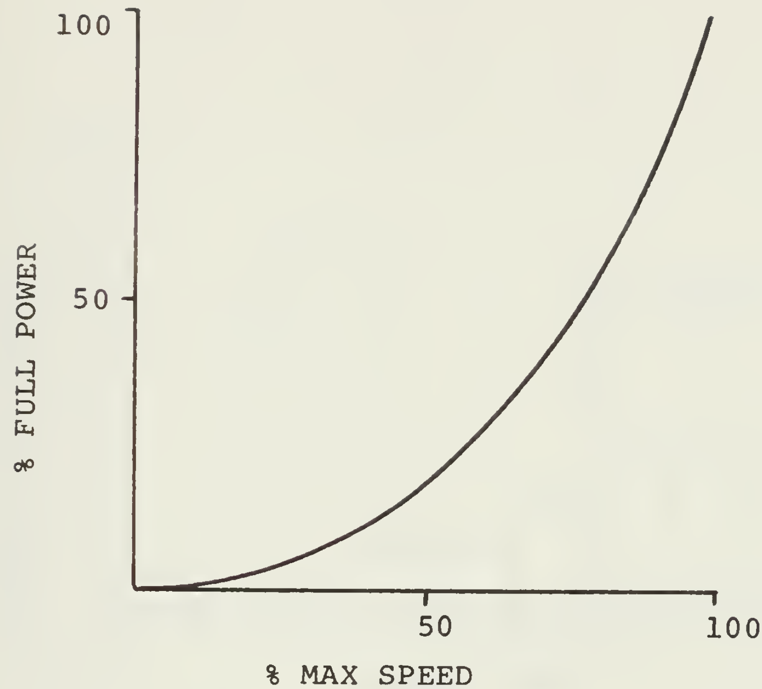


FIGURE 8 - TYPICAL MONOHULL POWERING CURVE

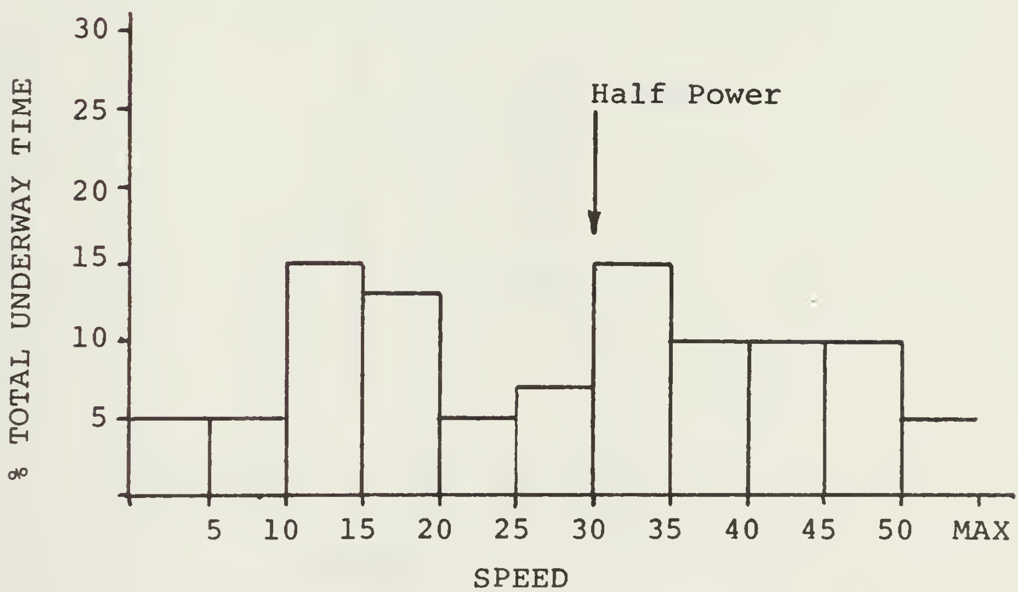


FIGURE 9 - TYPICAL HYDROFOIL SPEED - TIME DISTRIBUTION

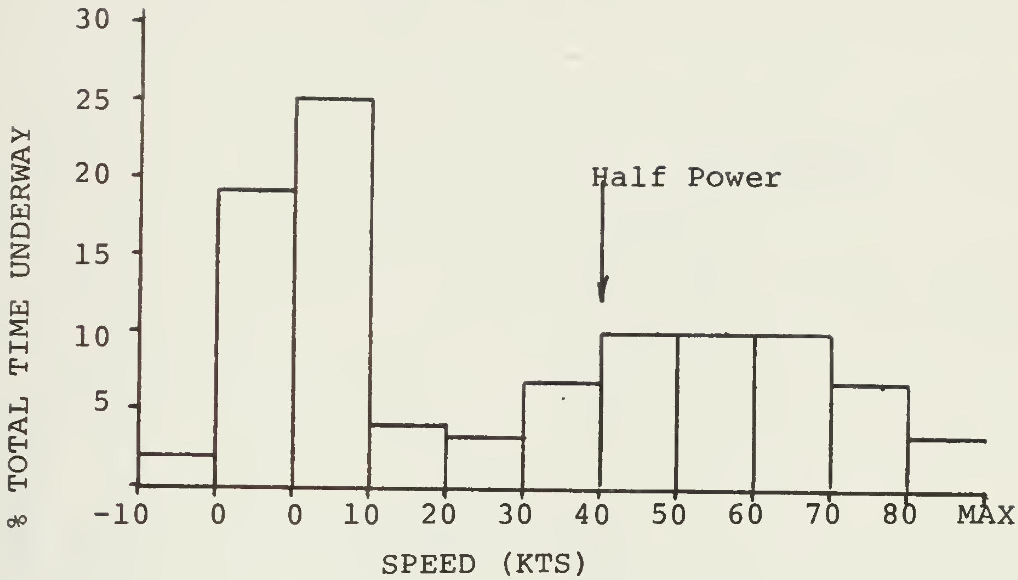


FIGURE 10 - TYPICAL SES SPEED - TIME DISTRIBUTION

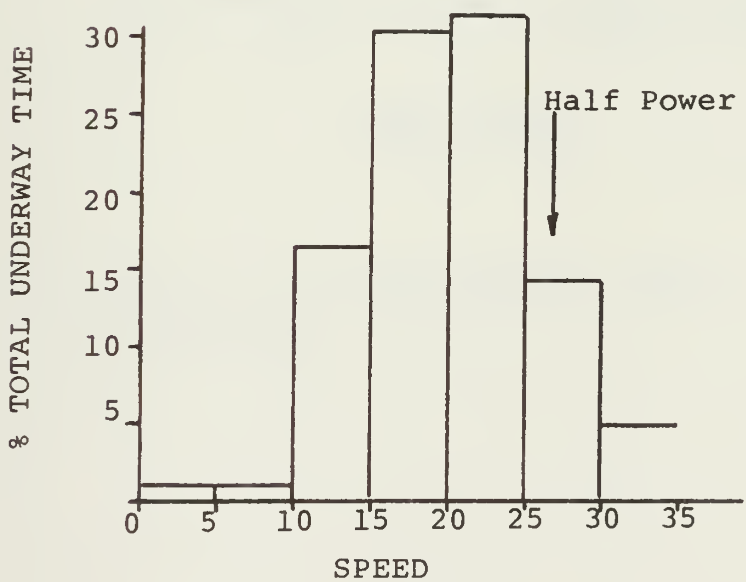


FIGURE 11 - TYPICAL DD SPEED - TIME DISTRIBUTION

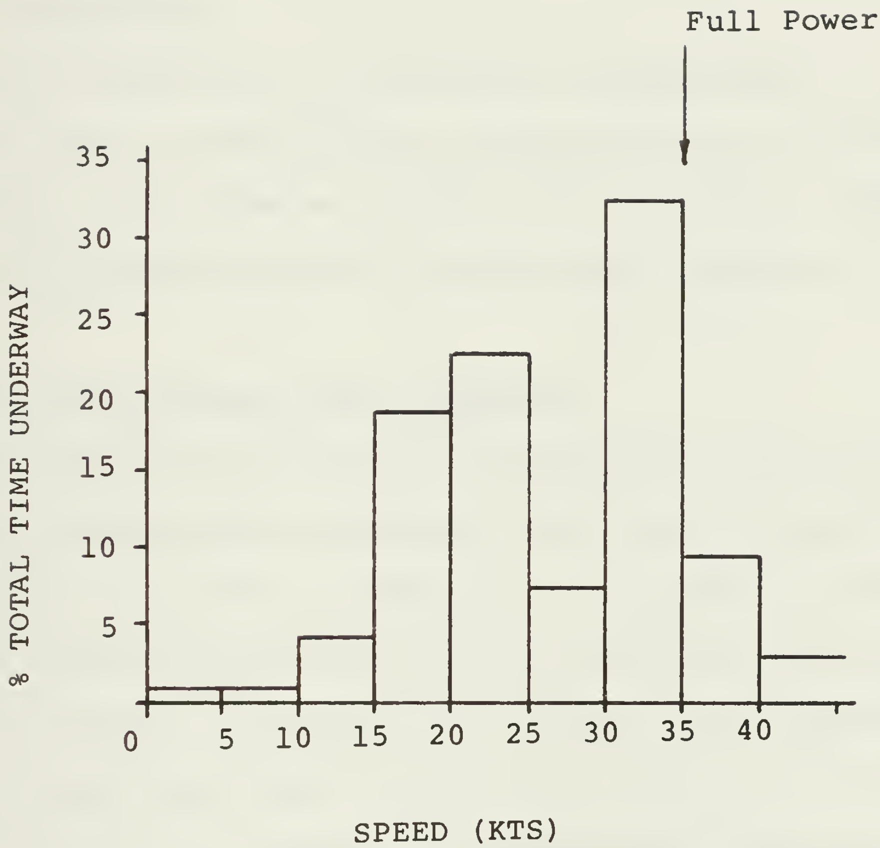


FIGURE 12 - TYPICAL HIGH SPEED MONOHULL SPEED - TIME DISTRIBUTION

A naval combatant must also be ready to engage threats and perform its designed missions at any random time. Full power availability is thus another measure of propulsion system operability.

In the remainder of this chapter comparisons of full and half power reliability and availability will be evaluated to determine the relative standing and reasons for differences between high performance and conventional propulsion systems.

6.2 Propulsion System Block Diagrams

Simplified block diagrams for high performance and conventional propulsion systems at full and half power are illustrated in Figures 13 through 21. The PHM and PG-84 have the same configuration for full and half power since both power levels are supplied by a single gas turbine.

All propulsion block diagrams are taken to the same level of detail and consist of the following subsystems:

--fuel oil service

--primemover

--speed reduction

--power transmission

--lubrication

--propulsor

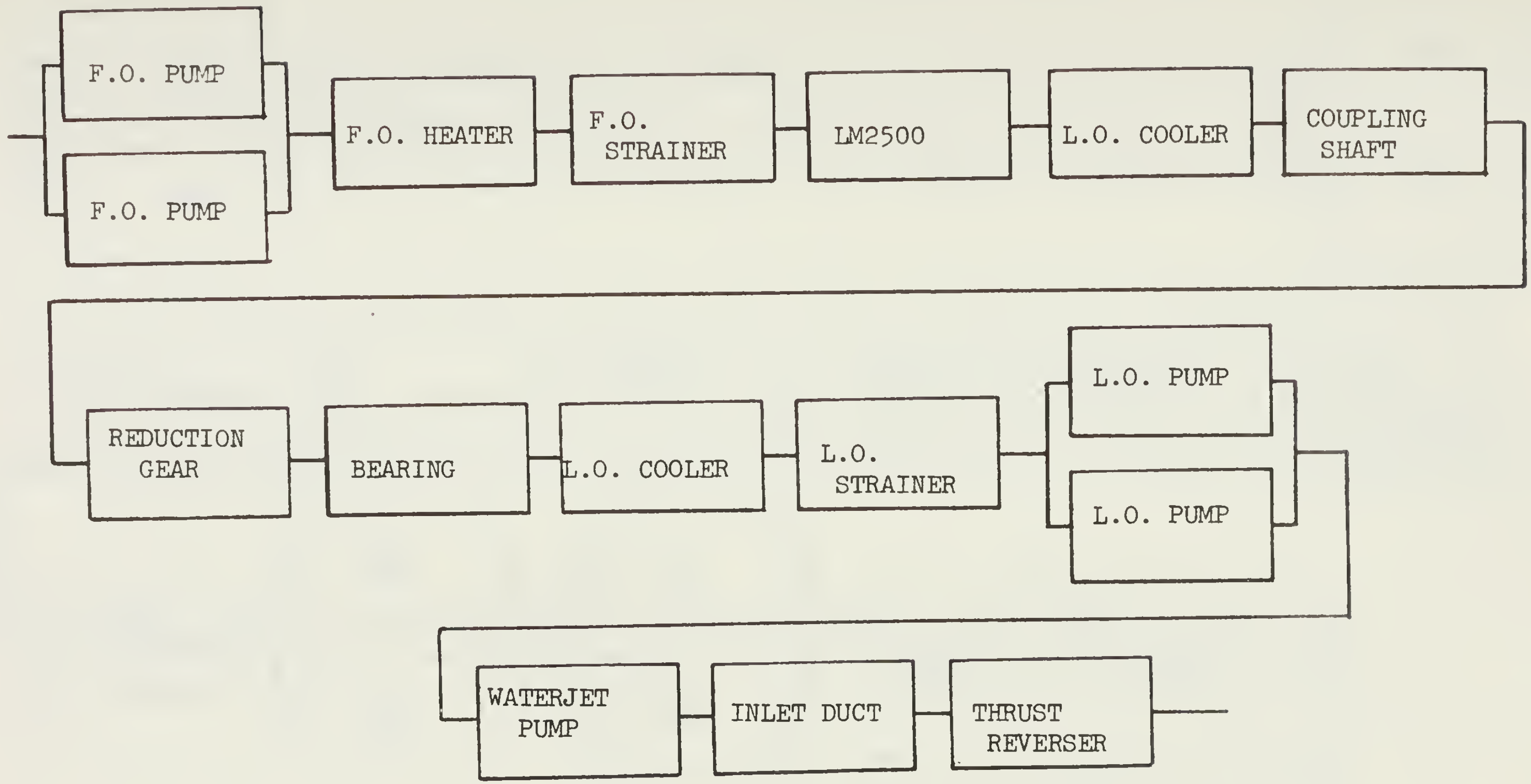


FIGURE 13 - PHM-1 PROPULSION BLOCK DIAGRAM - FULL POWER AND HALF POWER

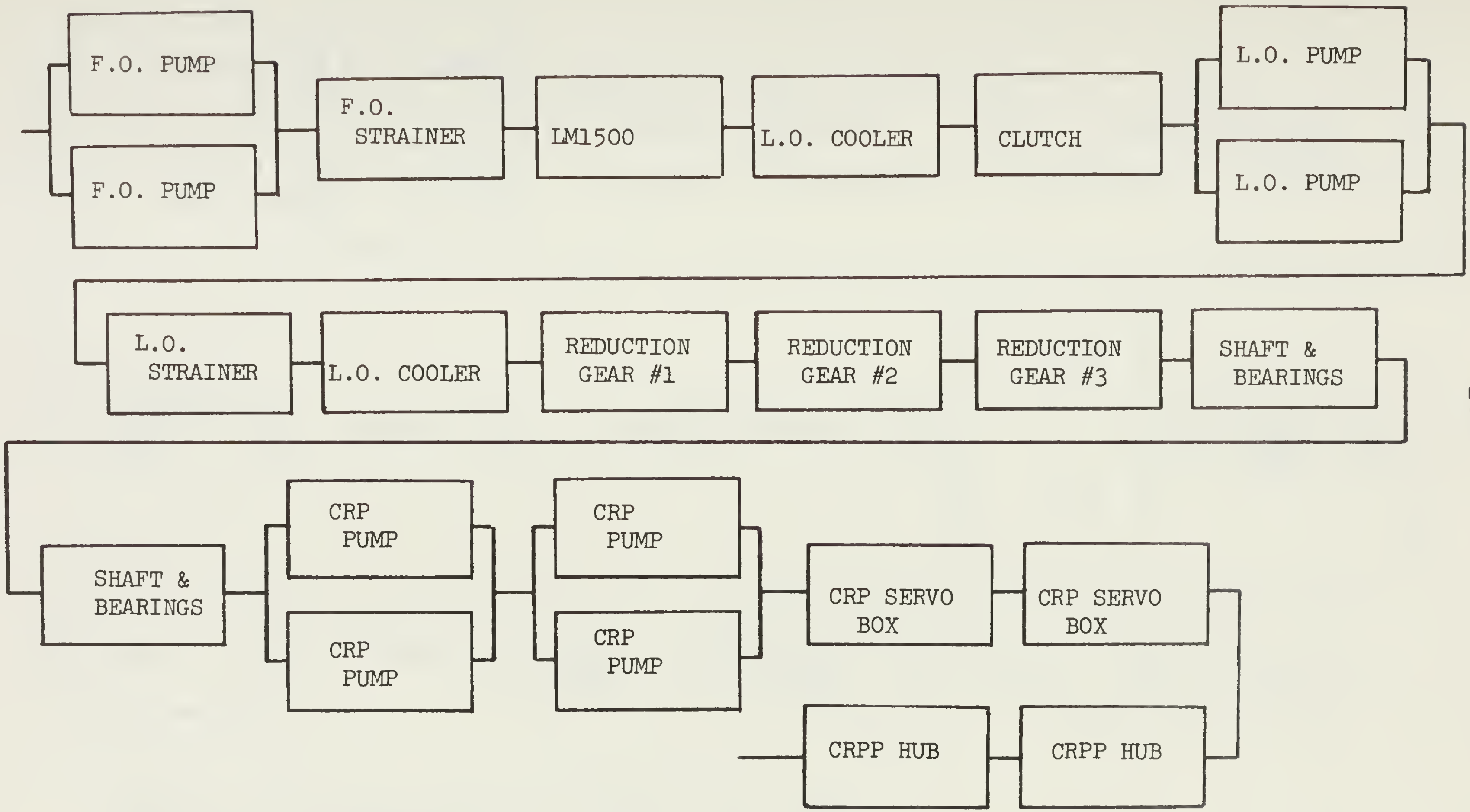


FIGURE 14 - PG-84 PROPULSION BLOCK DIAGRAM - FULL POWER AND HALF POWER

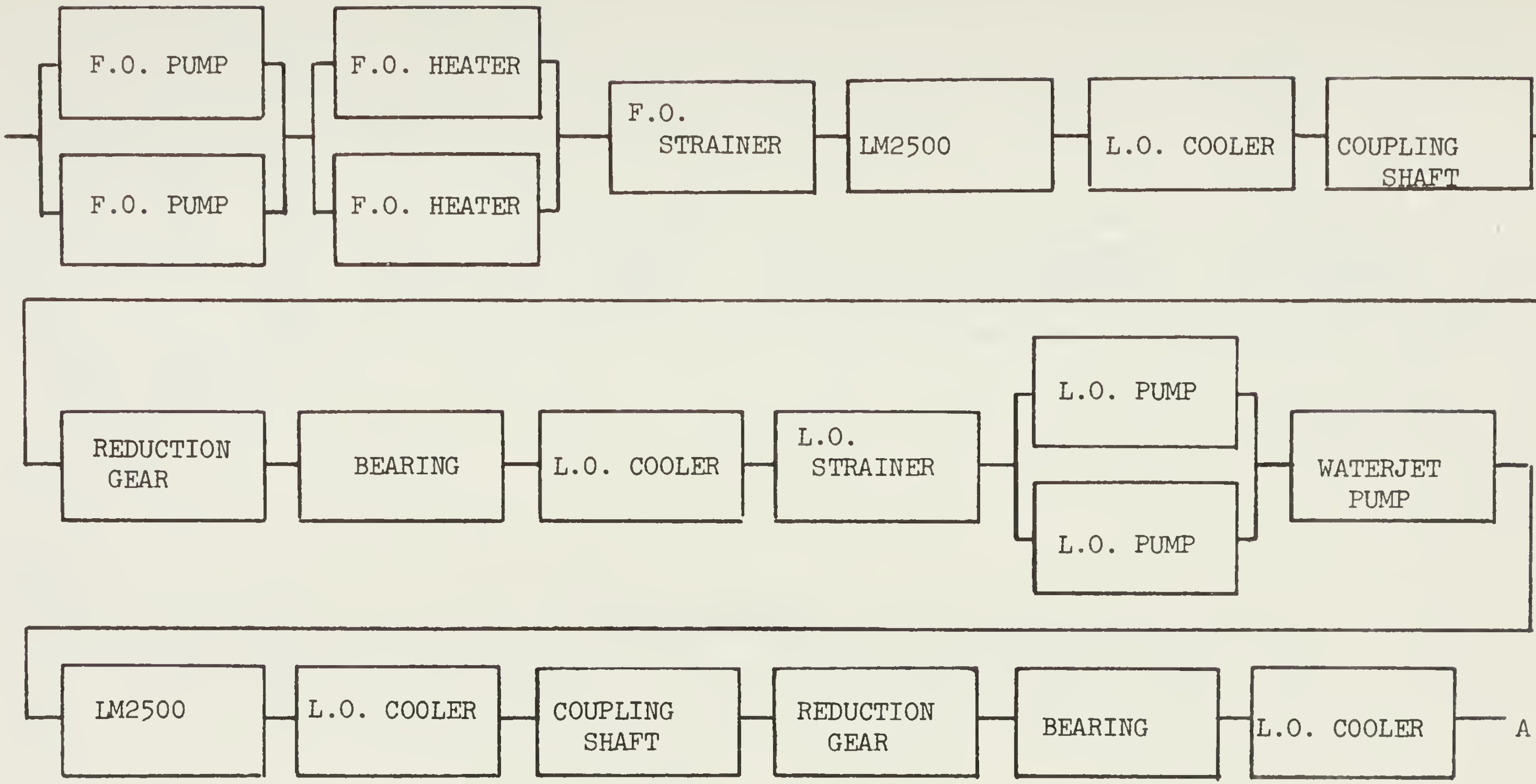
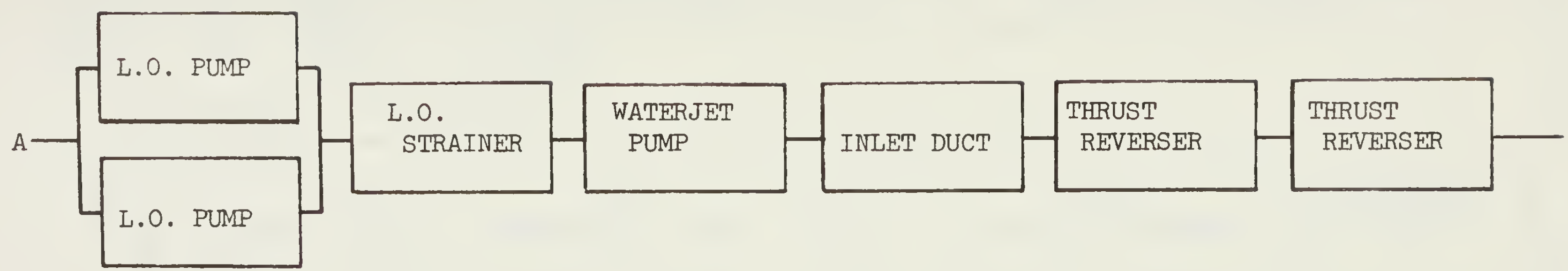


FIGURE 15 - LSES PROPULSION BLOCK DIAGRAM - FULL POWER



1 of 2 in Series

FIGURE 15 (continued) - LSES PROPULSION BLOCK DIAGRAM - FULL POWER

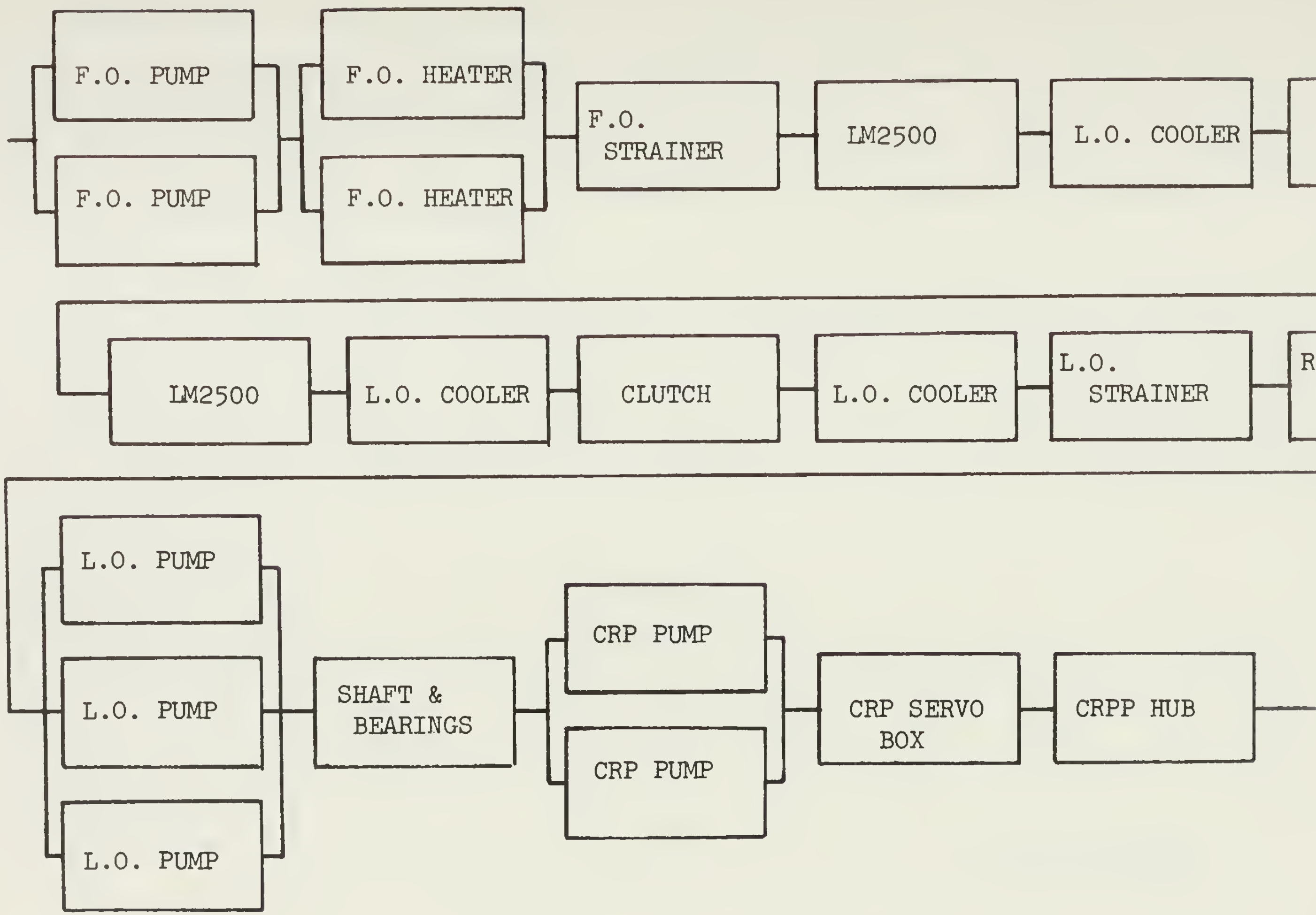


FIGURE 16 - FFG-7 PROPULSION BLOCK DIAGRAM-FULL POWER

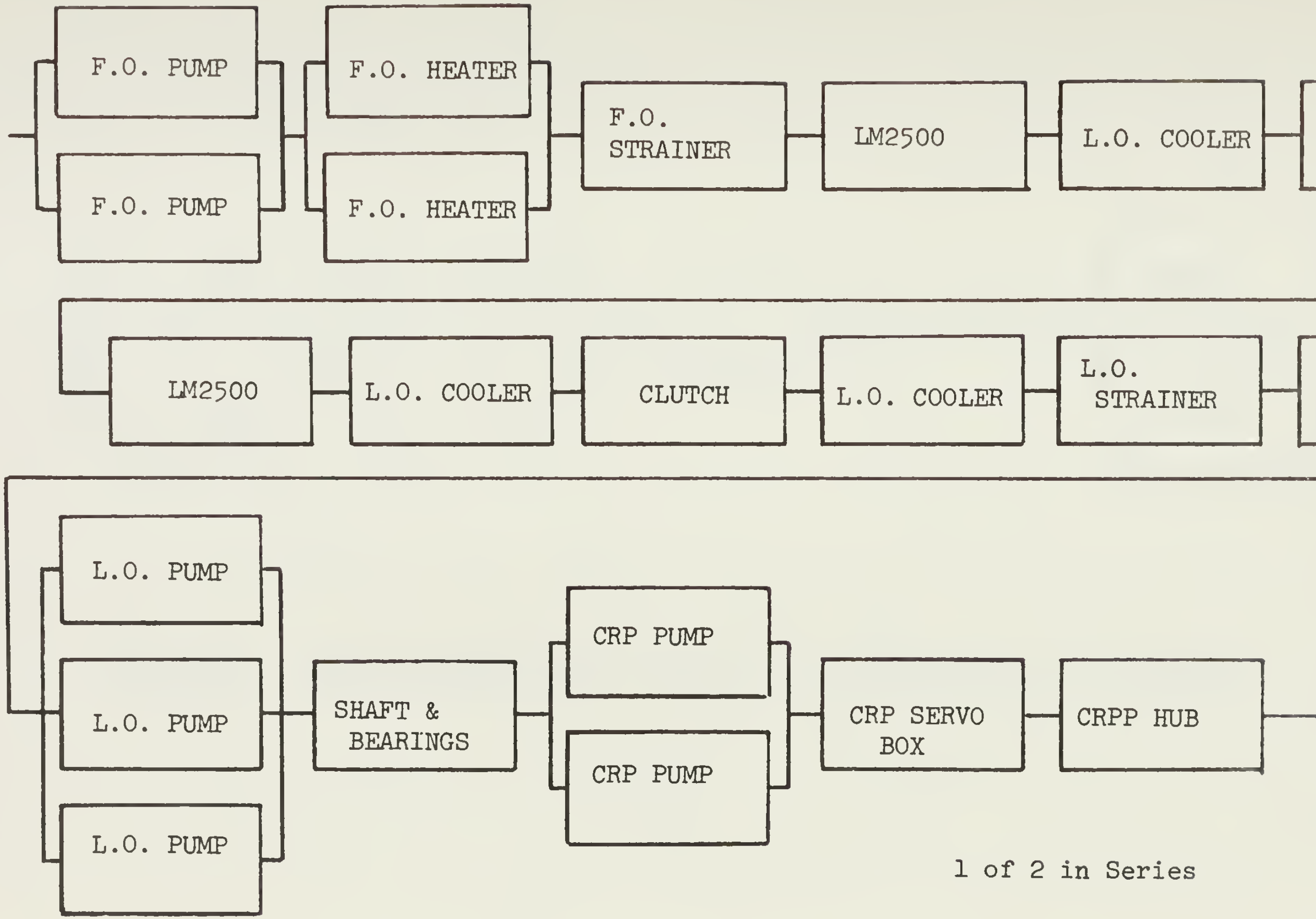


FIGURE 17 - DD-963 PROPULSION BLOCK DIAGRAM - FULL POWER

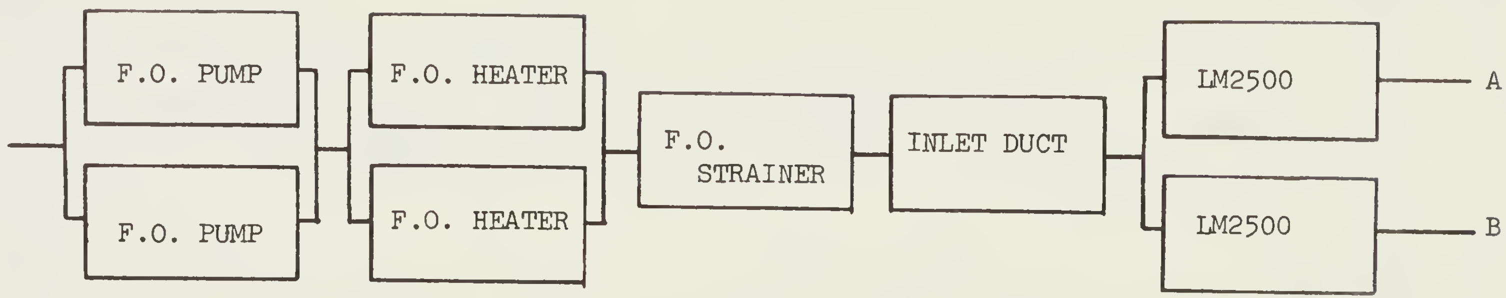
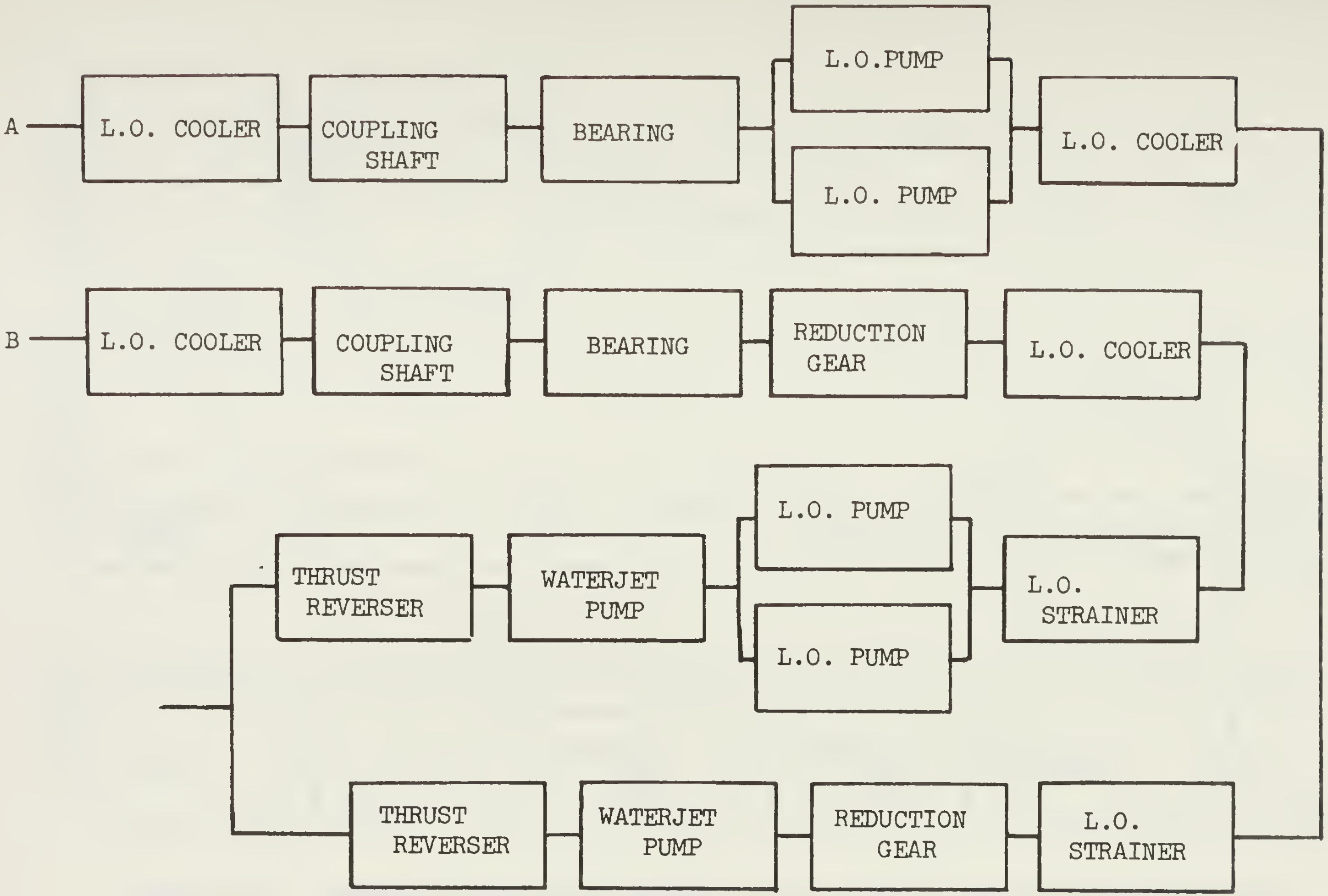


FIGURE 18a - LSES PROPULSION BLOCK DIAGRAM - HALF POWER-SERIES CONFIGURATION



1 of 2 in Series

FIGURE 18a (continued) - LSES PROPULSION BLOCK DIAGRAM-HALF POWER - SERIES CONFIGURATION

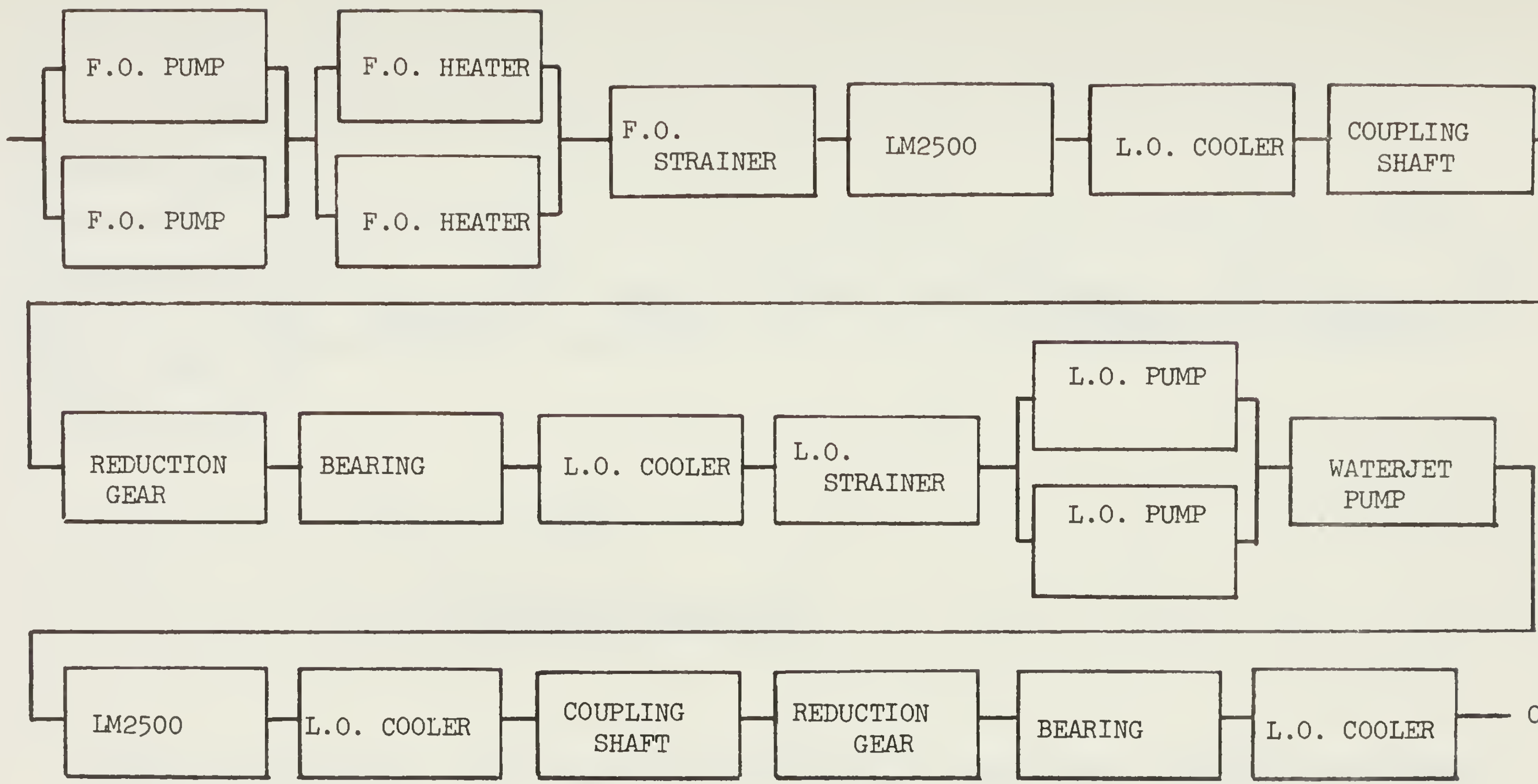
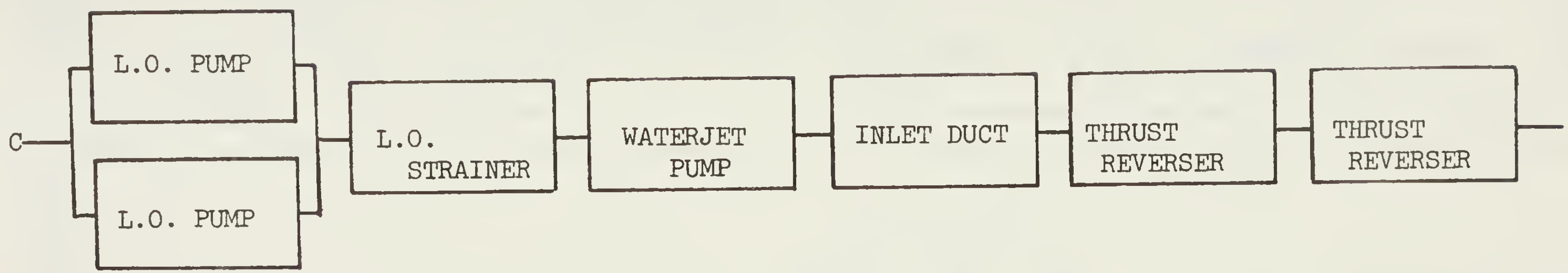


FIGURE 18b - LSES PROPULSION BLOCK DIAGRAM - HALF POWER - PARALLEL CONFIGURATION



1 of 2 in Parallel

$$R = 1-(1-R^2_{series})(1-R_{parallel})^2$$

FIGURE 18b (continued) - LSES PROPULSION BLOCK DIAGRAM - HALF POWER - PARALLEL CONFIGURATION

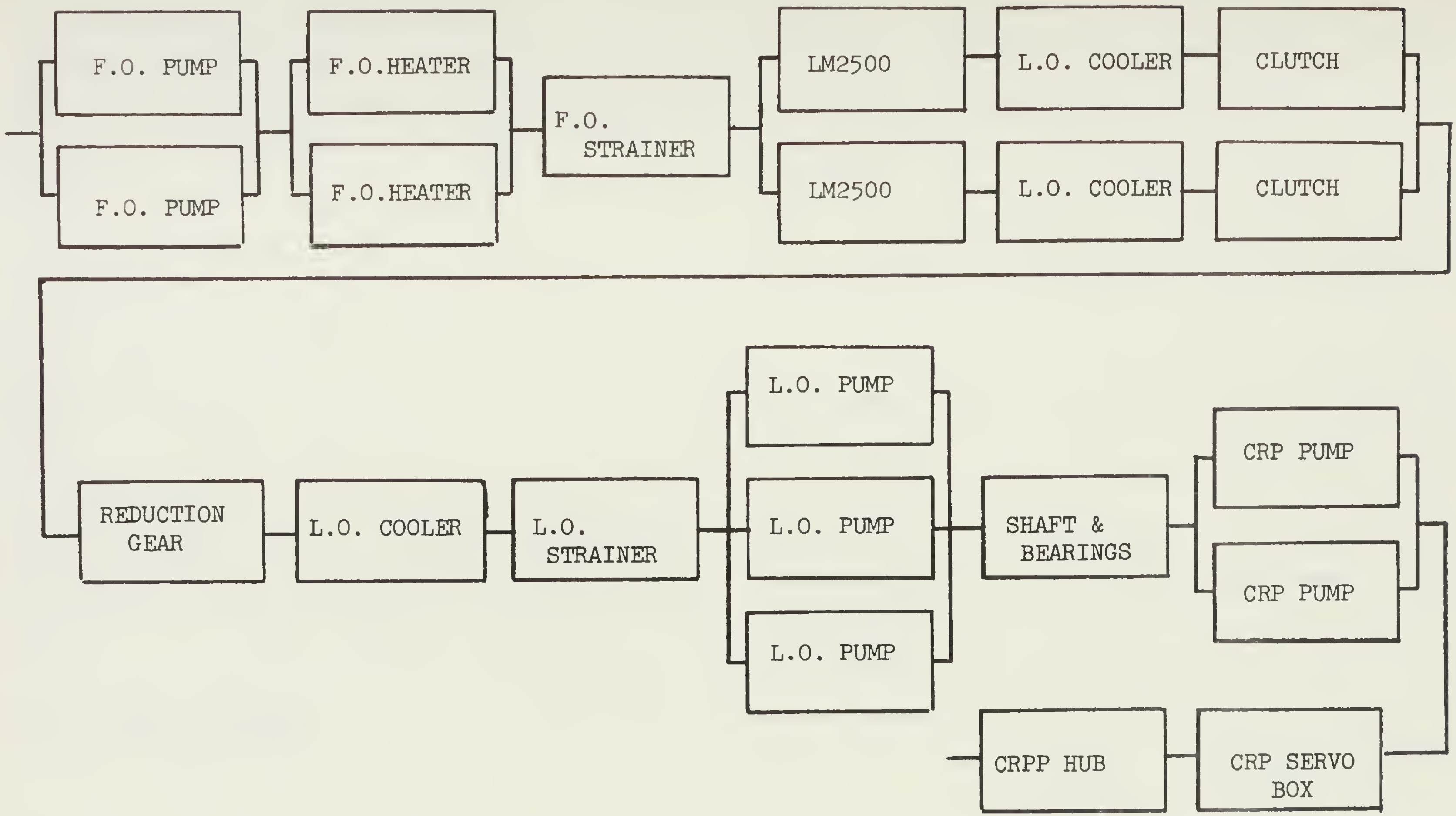
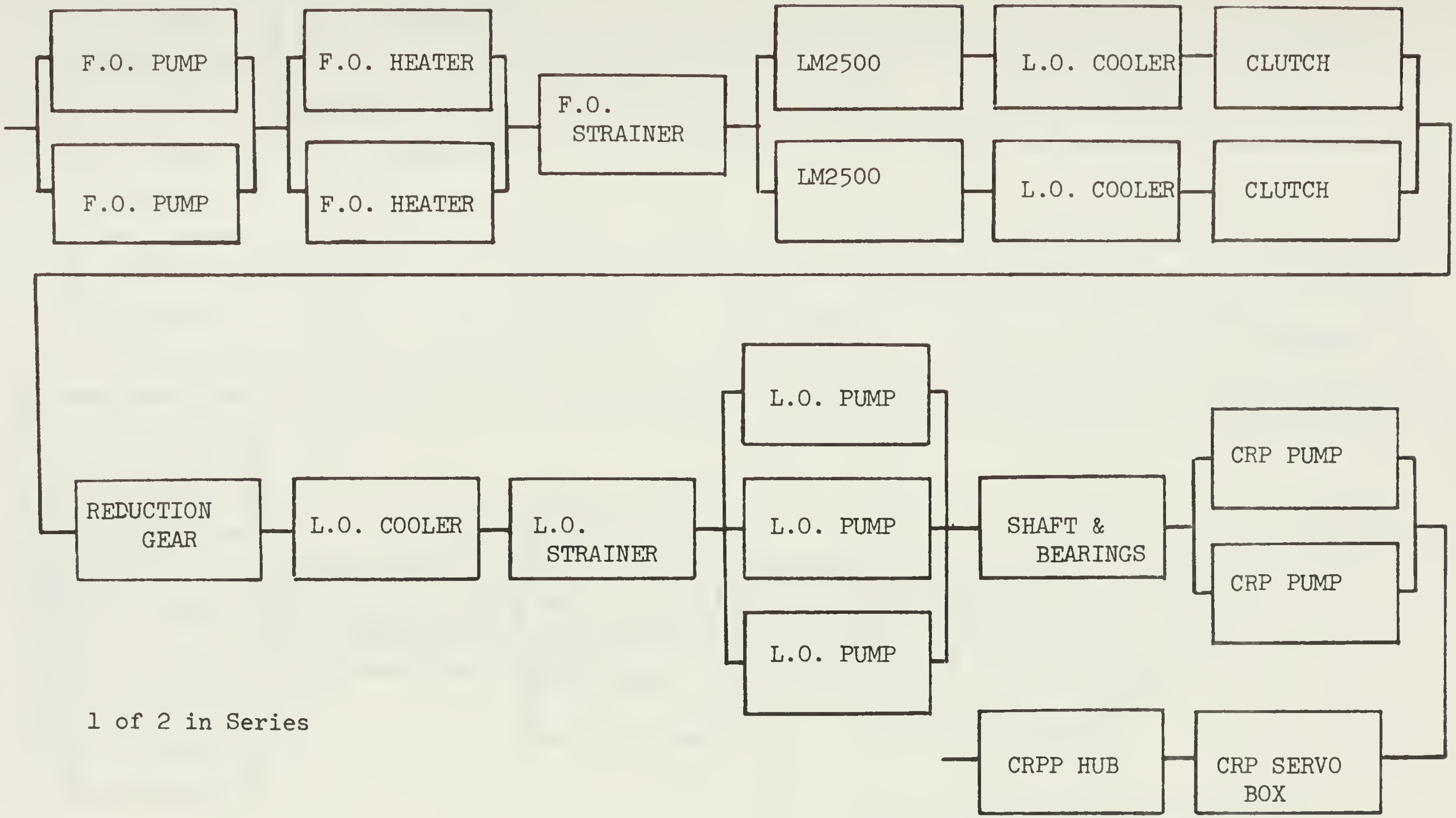
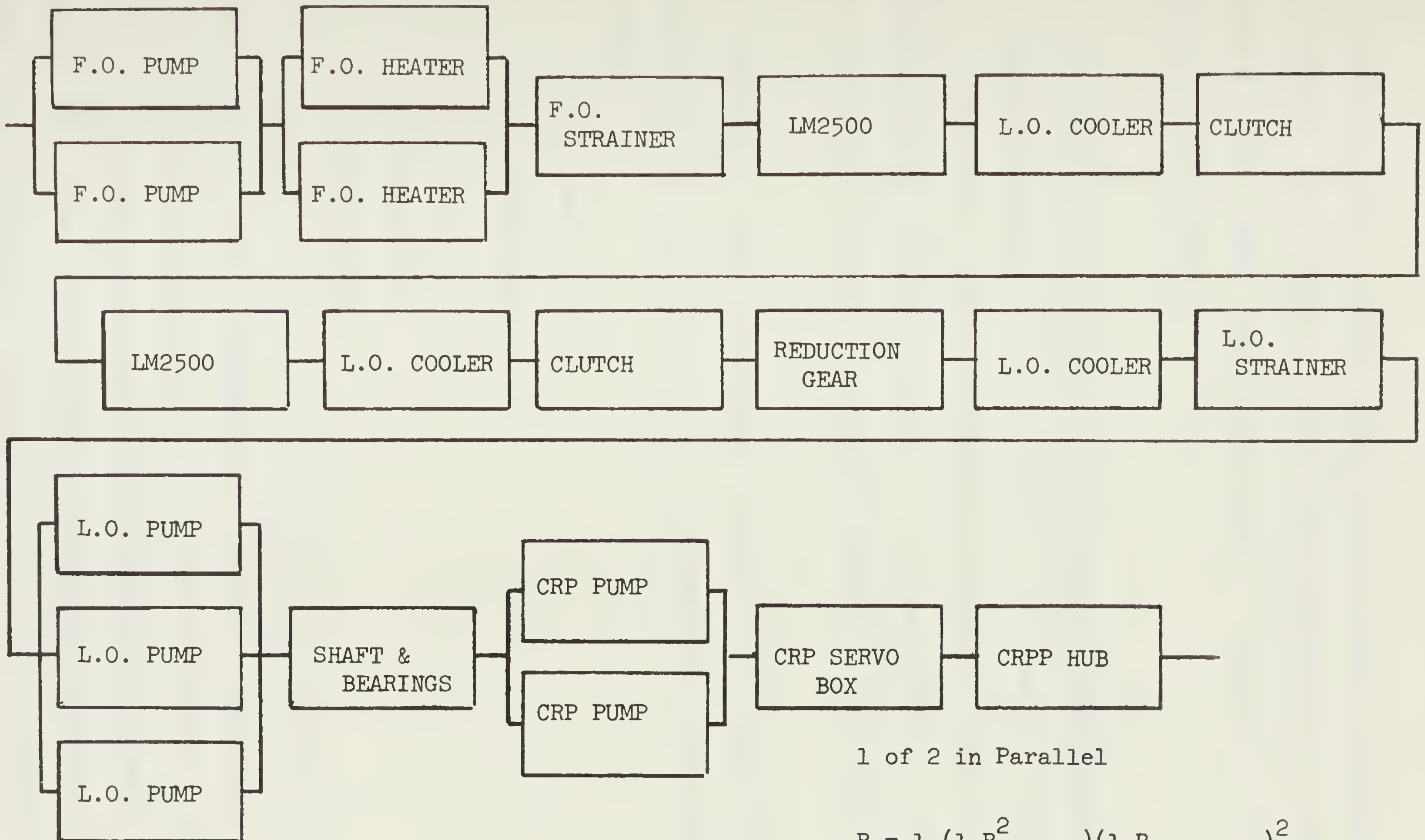


FIGURE 19 - FFG-7 PROPULSION BLOCK DIAGRAM - HALF POWER



1 of 2 in Series

FIGURE 20 - DD-963 PROPULSION BLOCK DIAGRAM - HALF POWER - SERIES CONFIGURATION



$$R = 1-(1-R^2_{series})(1-R_{parallel})^2$$

FIGURE 21 - DD-963 PROPULSION BLOCK DIAGRAM - HALF POWER - PARALLEL CONFIGURATION

For the larger ships, there are multiple configurations that produce half power, so two block diagrams are presented for half power along with the method for accounting for this additional redundancy.

6.3 R/M/A Resulting from Configuration and Redundancy

The reliability and availability inherent in each propulsion system resulting from its configuration and redundancy will be investigated. The assumption is made that all generically similar components exhibit the same MTBF and MTTR.

Several of the components installed in these propulsion systems are not repairable while underway by the crew. They include:

--gas turbines

--clutches

--reduction gear

--shafting

--propulsors

This analysis uses the principles discussed in Chapter 3 concerning off-line redundancy of standby components.

Since the mission durations vary from ship to ship and it is of interest to compare the propulsion system on an equal basis, the reliabilities and availabilities of each propulsion system are calculated for mission durations of

5, 15, and 30 days. Table 13 lists the MTBF and MTTR data used in the analysis.

The results are plotted on Figures 22 through 26 and illustrate the reduction of reliability and availability with time. Full power reliability and availability decrease as installed shaft horsepower increases as shown in Figures 22 and 23. This is due to the increased number of components necessary to achieve the required power level. If large powering requirements could be met with one primemover and propulsor combination, the high SHP systems could be as reliable and available as low SHP ships.

The differences between PHM-1 and PG-84, and LSES and DD-963 reliabilities and availabilities are a result of the different numbers of propulsors in otherwise similar configurations.

The similarity in reliabilities and availabilities for PHM and PG-84, and LSES and DD-963 illustrated in Figures 22 and 26 indicates that there exists a similarity in configuration redundancy of these ship pairs. Comparing Figure 13 with Figure 14 and Figure 15 with Figure 17, an equivalent degree of redundancy is observed between high performance and conventional displacement propulsion systems.

The improvement of half power availability and reliability with installed SHP is illustrated in Figures 24 and 25. Good half power reliability and availability

TABLE 13

MTBF/MTTR DATA - CONFIGURATION

Component	MTBF	MTTR	R(5)	R(15)	R(30)	A
F.O.S.P.	40,000	6.2	0.9970	0.9910	0.9821	0.9998
F.O. HTR	333,330	1.2	0.9996	0.9989	0.9978	0.9999
GAS TURBINE	6,450	NR	0.9815	0.9457	0.8943	--
L.O. COOLER	90,576	3	0.9986	0.9960	0.9920	0.9999
CLUTCH	50,000	NR	0.9976	0.9928	0.9857	--
REDUCTION GEAR	200,000	NR	0.9994	0.9982	0.9964	--
L.O. PUMP	4,000	5	0.9700	0.9139	0.8352	0.9987
SHAFT/ BEARINGS	200,000	NR	0.9994	0.9982	0.9964	--
F.O. STRAINER	60,000	3	0.9980	0.9940	0.9880	0.9999
L.O. STRAINER	60,000	3	0.9980	0.9940	0.9880	0.9999
DIESEL	3,000	8	0.9607	0.8869	0.7866	0.9973
PROPULSOR	100,000	NR	0.9988	0.9964	0.9928	--

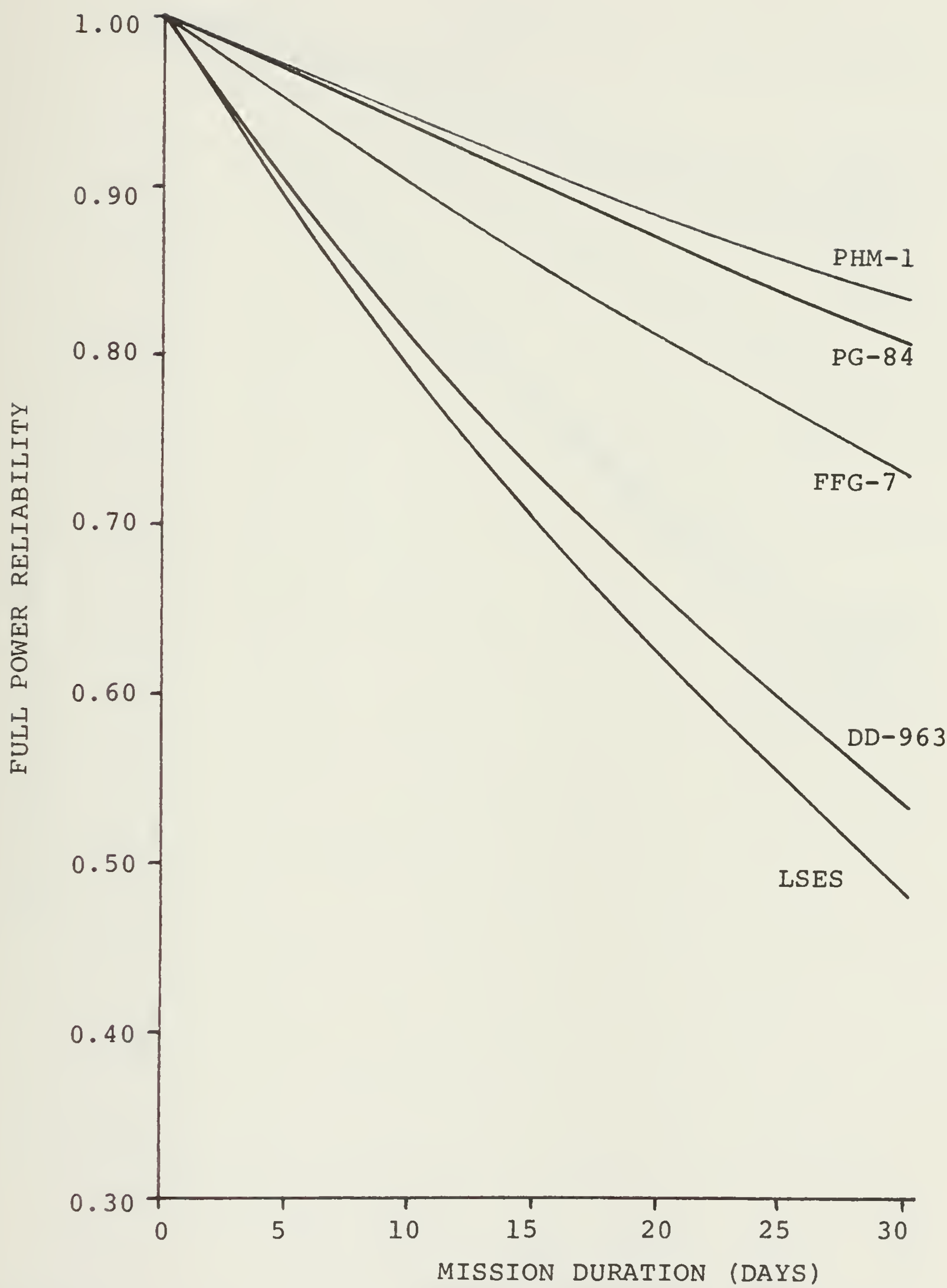


FIGURE 22 - FULL POWER RELIABILITY VS. MISSION DURATION
CONFIGURATION EFFECTS

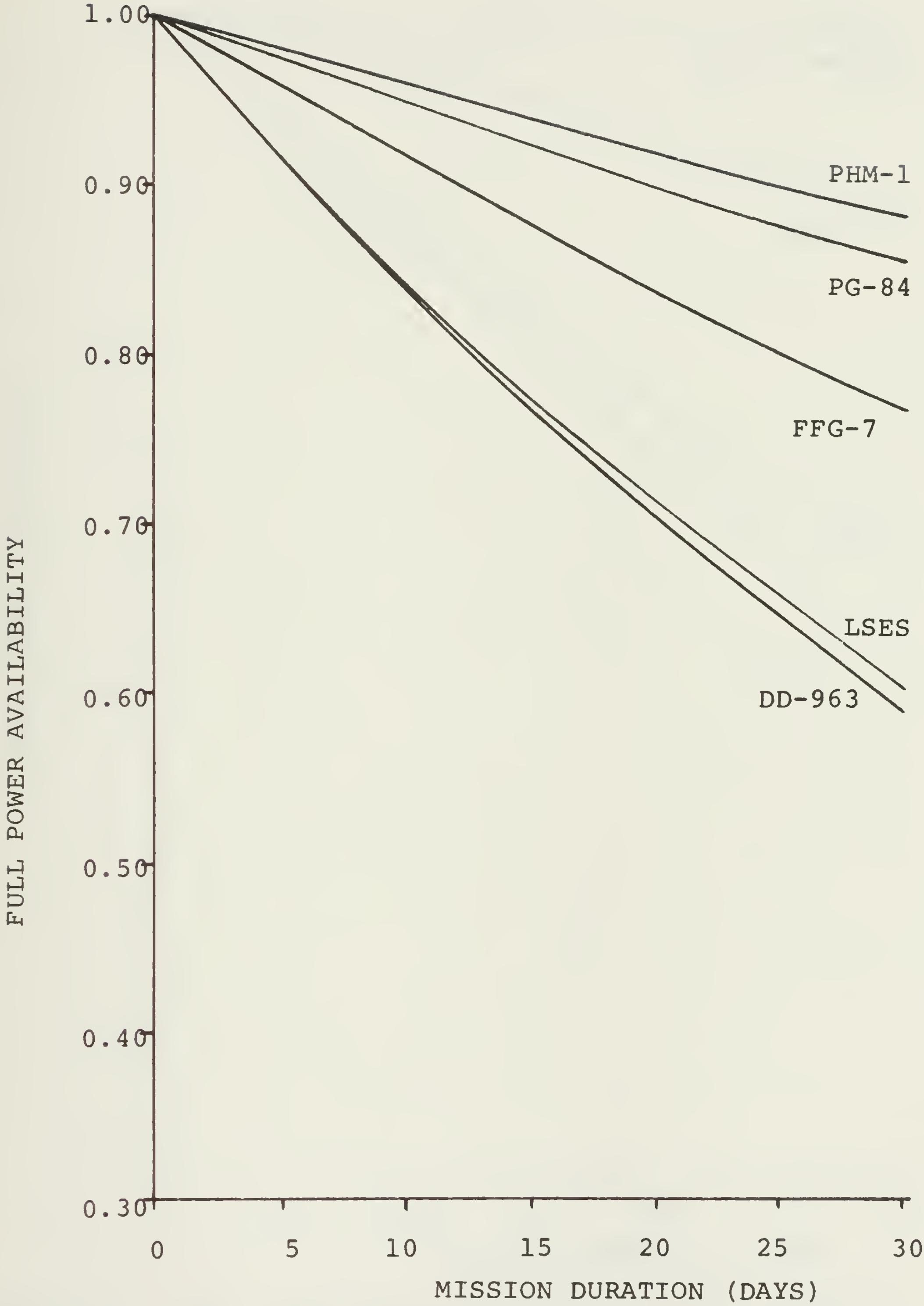


FIGURE 23 - FULL POWER AVAILABILITY VS. MISSION DURATION
CONFIGURATION EFFECTS

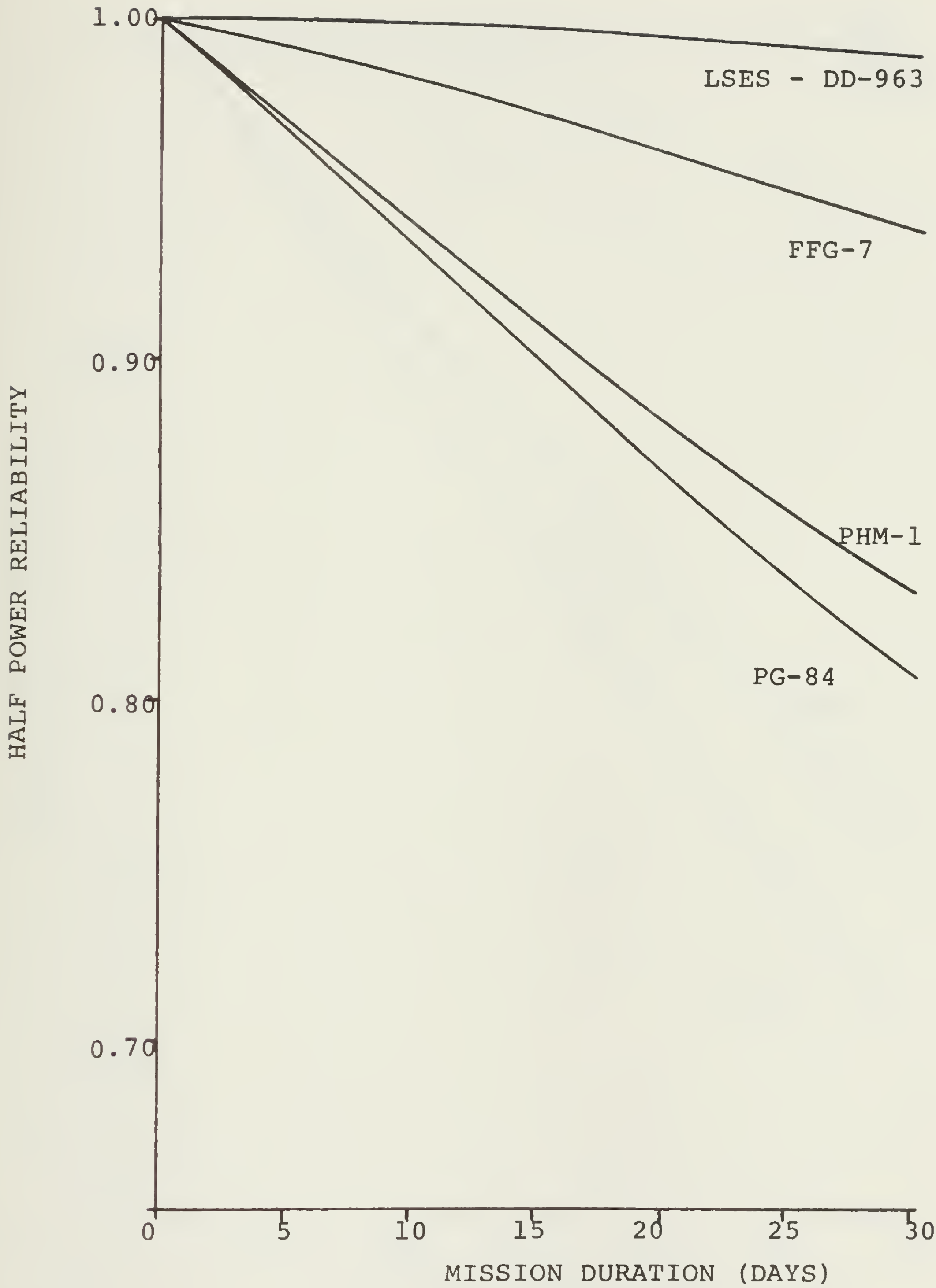


FIGURE 24 - HALF POWER RELIABILITY VS. MISSION DURATION
CONFIGURATION EFFECTS

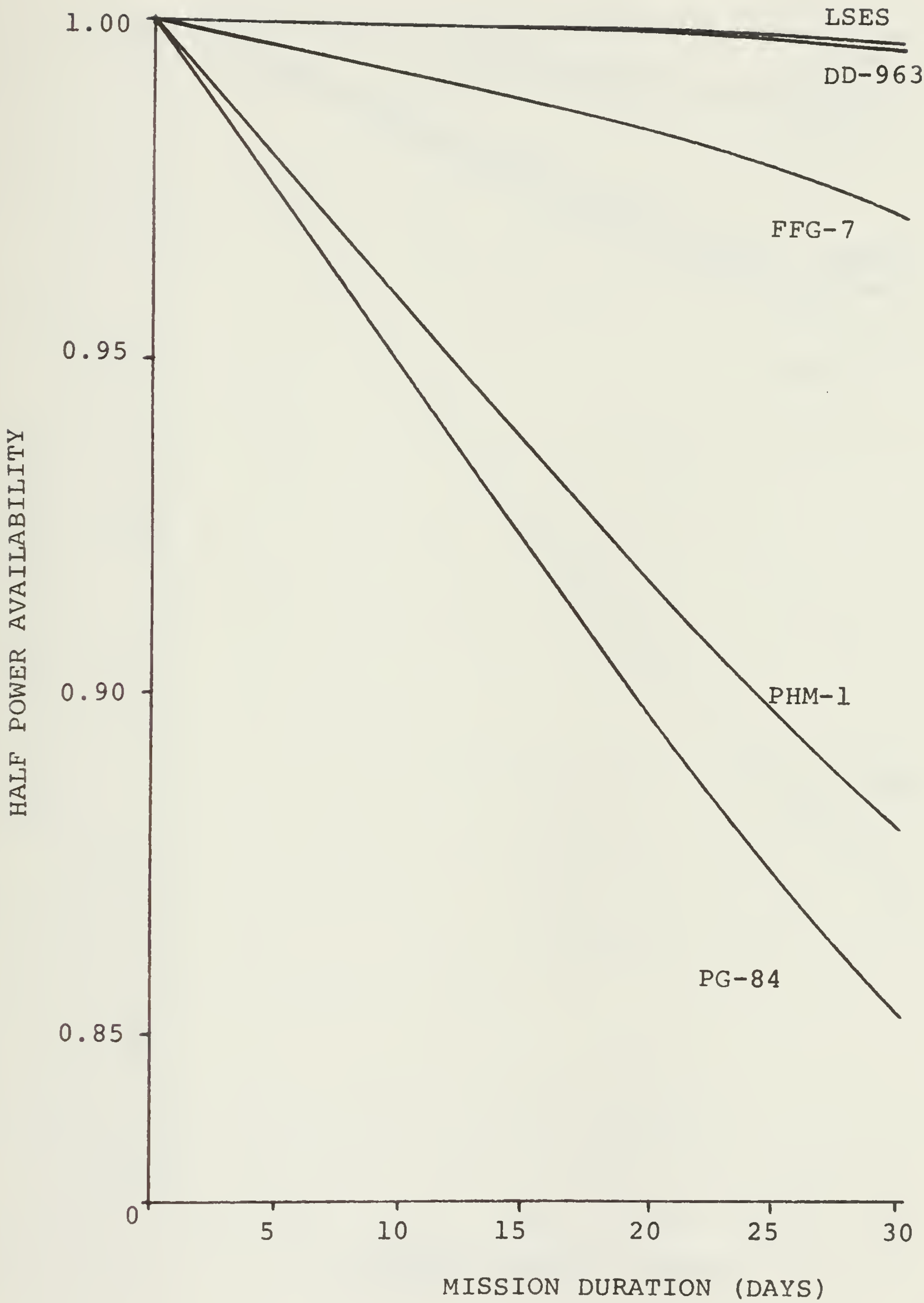


FIGURE 25 - HALF POWER AVAILABILITY VS. MISSION DURATION CONFIGURATION EFFECTS

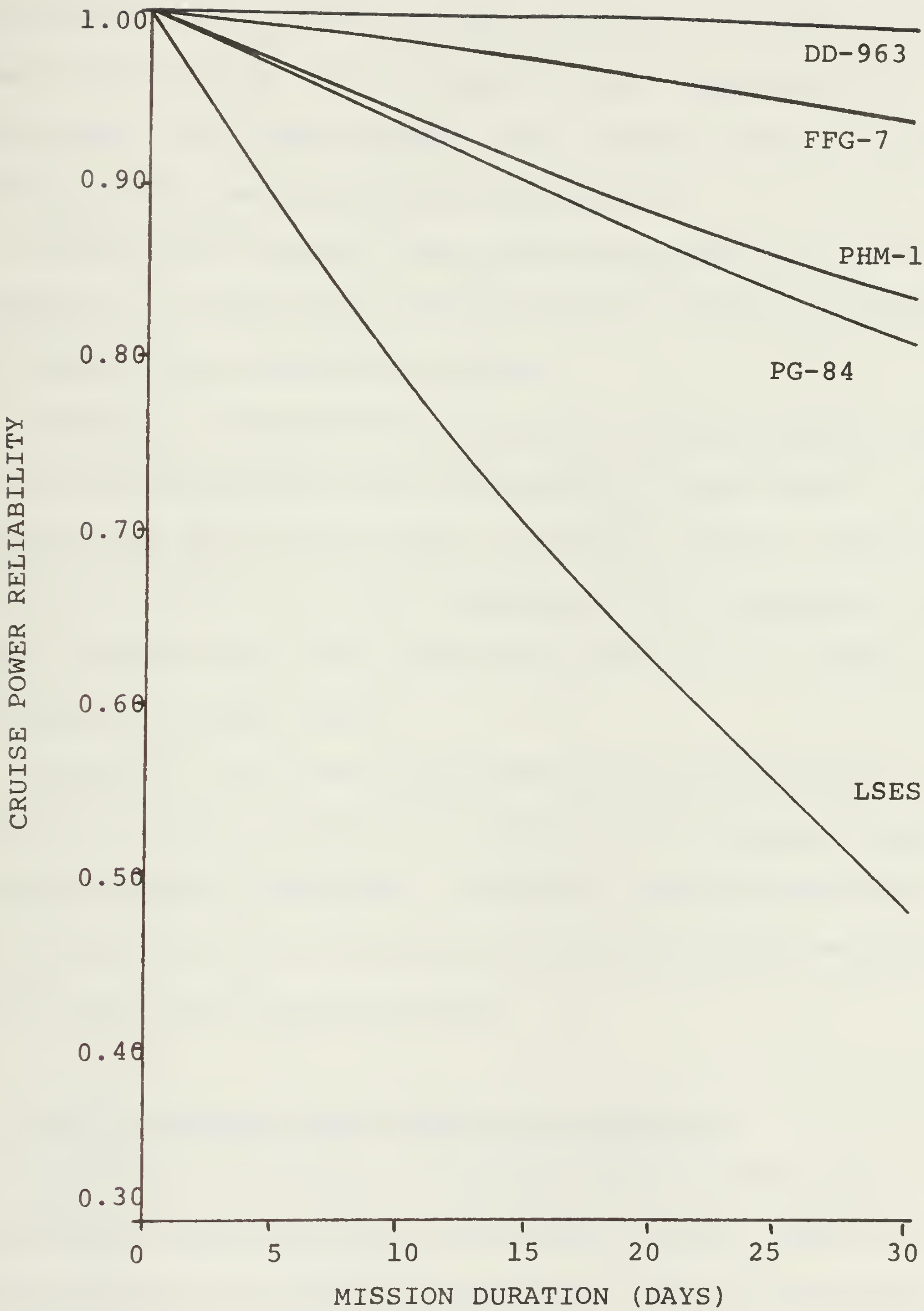


FIGURE 26 - CRUISE POWER RELIABILITY VS. MISSION DURATION
CONFIGURATION EFFECTS

are derived from the use of many primemovers. This is in direct conflict with good reliability and availability at full power. The ship designer must evaluate which of these configurations has priority in the design.

As with full power, there exist equivalent levels of redundancy in half power configurations of high performance and conventional propulsion systems.

Figure 26 presents the propulsion system reliabilities for the configurations most meaningfull to each ship. Thus PHM and LSES are at full power, and PG-84, FFG-7, and DD-963 are at half power. For the purposes of this analysis, the power level at which the ship spends most of its time will be defined as "cruise" power.

Generally, the high performance ships will be less reliable in this cruise power configuration because there is no primemover redundancy. PG-84 is less reliable than PHM, however, due to its dual screw configuration as compared to the PHM single waterjet pump.

6.4 R/M/A Resulting from Types of Propulsors

The major difference in configuration between high performance and conventional propulsion systems used in this analysis is the type of propulsor. High performance propulsion systems use low weight and volume waterjet pumps

and reduction gear boxes. Conventional displacement propulsion systems use heavy controllable reversable pitch propellers and reduction gear.

In operation, the waterjet pumps have poorer MTBF's than CRP propellers. This should result in high performance propulsion system being less reliable and available than predicted in the previous section. To evaluate the reduction in reliability and availability caused by these propulsors, a similar analysis to that done in Section 6.3 will be conducted with the only exception being difference in the MTBF and MTTR data used for the propulsors. Table 14 lists the MBTF and MTTR data to be used.

In Figures 27 and 29 the reliabilities of the high performance propulsion systems fall well below those of the convention systems as would be expected. The availabilities of high performance propulsion systems, however, do not degrade below those of their conventional displacement counterparts. This is due to the larger number of components used in the CRP propeller subsystem which degrades conventional propulsion system availability at about the same rate as the simple waterjet pump propulsors degrade the high performance system.

Therefore, the type of propulsor significantly effects the reliability of a propulsion system. The waterjet pump propulsor reduces high performance propulsion system

TABLE 14

MTBF/MTTR DATA - PROPULSOR TYPE

Component	MTBF	MTTR	R(5)	R(15)	R(30)	A
F.O.S.P.	40,000	6.2	0.9970	0.9910	0.9821	0.9998
F.O. HEATER	333,330	1.2	0.9996	0.9989	0.9978	0.9999
LM2500	6,450	NR	0.9815	0.9457	0.8943	--
LM1500	6,450	NR	0.9815	0.9457	0.8943	--
L.O. COOLER	90,576	3	0.9986	0.9960	0.9920	0.9999
CLUTCH	50,000	NR	0.9976	0.9928	0.9857	--
REDUCTION GEAR	200,000	NR	0.9994	0.9982	0.9964	--
L. O. PUMP	4,000	5	0.9700	0.9139	0.8352	0.9987
SHAFT/BRG	200,000	NR	0.9994	0.9982	0.9964	--
CRP PUMP	100,000	2	0.9988	0.9964	0.9928	0.9999
CRP SERVO BOX	333,000	9	0.9996	0.9989	0.9978	0.9999
CRP HUB	100,000	NR	0.9988	0.9964	0.9928	--
F.O. STRAINER	60,000	3	0.9980	0.9940	0.9880	0.9999
L.O. STRAINER	60,000	3	0.9980	0.9940	0.9880	0.9999
PROPELLER	200,000	NR	0.9994	0.9982	0.9964	--
CPLG SHAFT	72,780	4	0.9983	0.9950	0.9901	0.9999
WATERJET PUMP	6,700	8	0.9822	0.9476	0.8981	0.9988
DIESEL	3,000	8	0.9607	0.8869	0.7866	0.9973
INLET DUCT	45,000	NR	0.9973	0.9920	0.9841	--
THRUST REVERSER	6,150	6	0.9807	0.9431	0.8895	0.9990

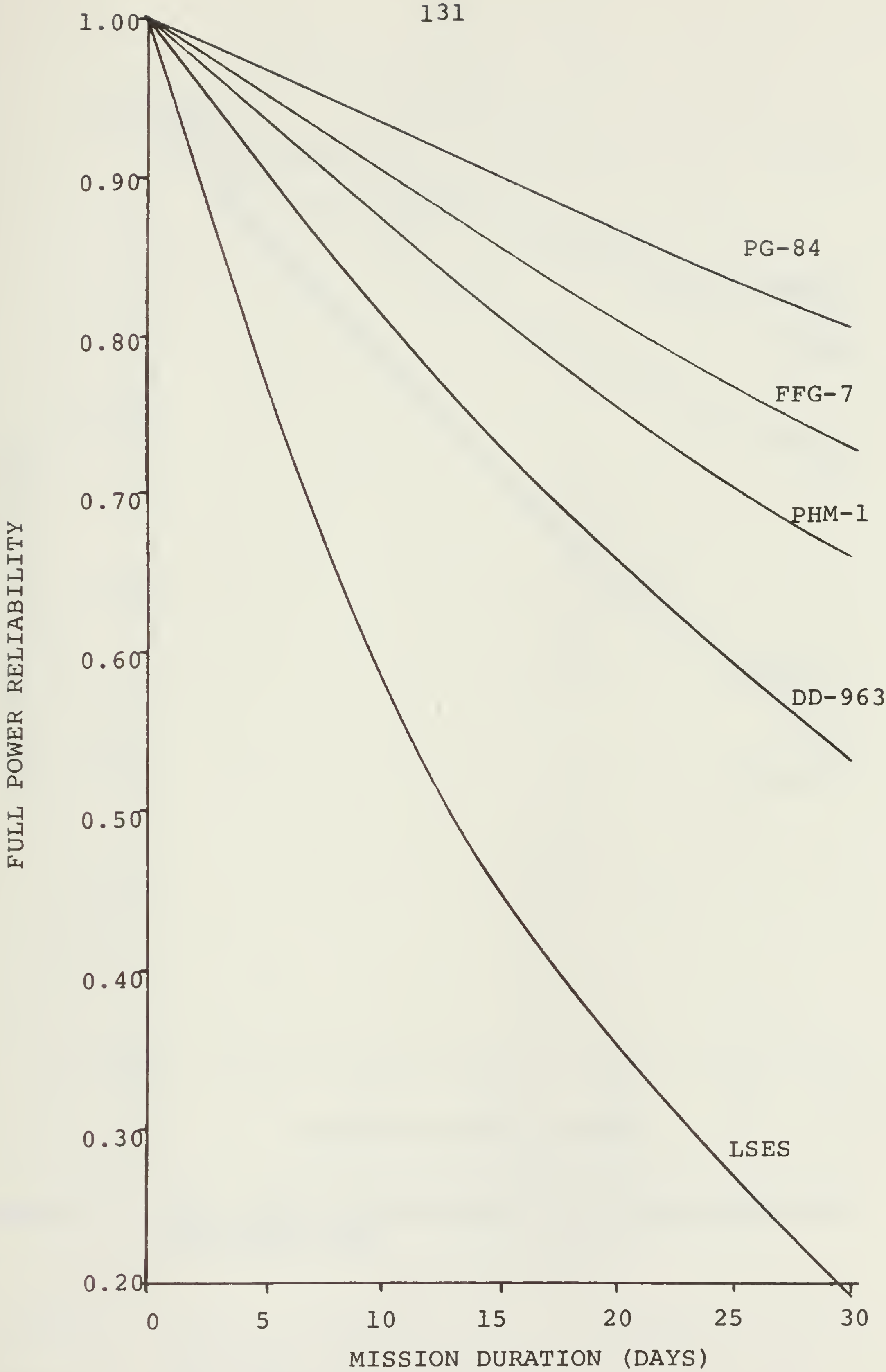


FIGURE 27 - FULL POWER RELIABILITY VS. MISSION DURATION
PROPULSOR TYPE

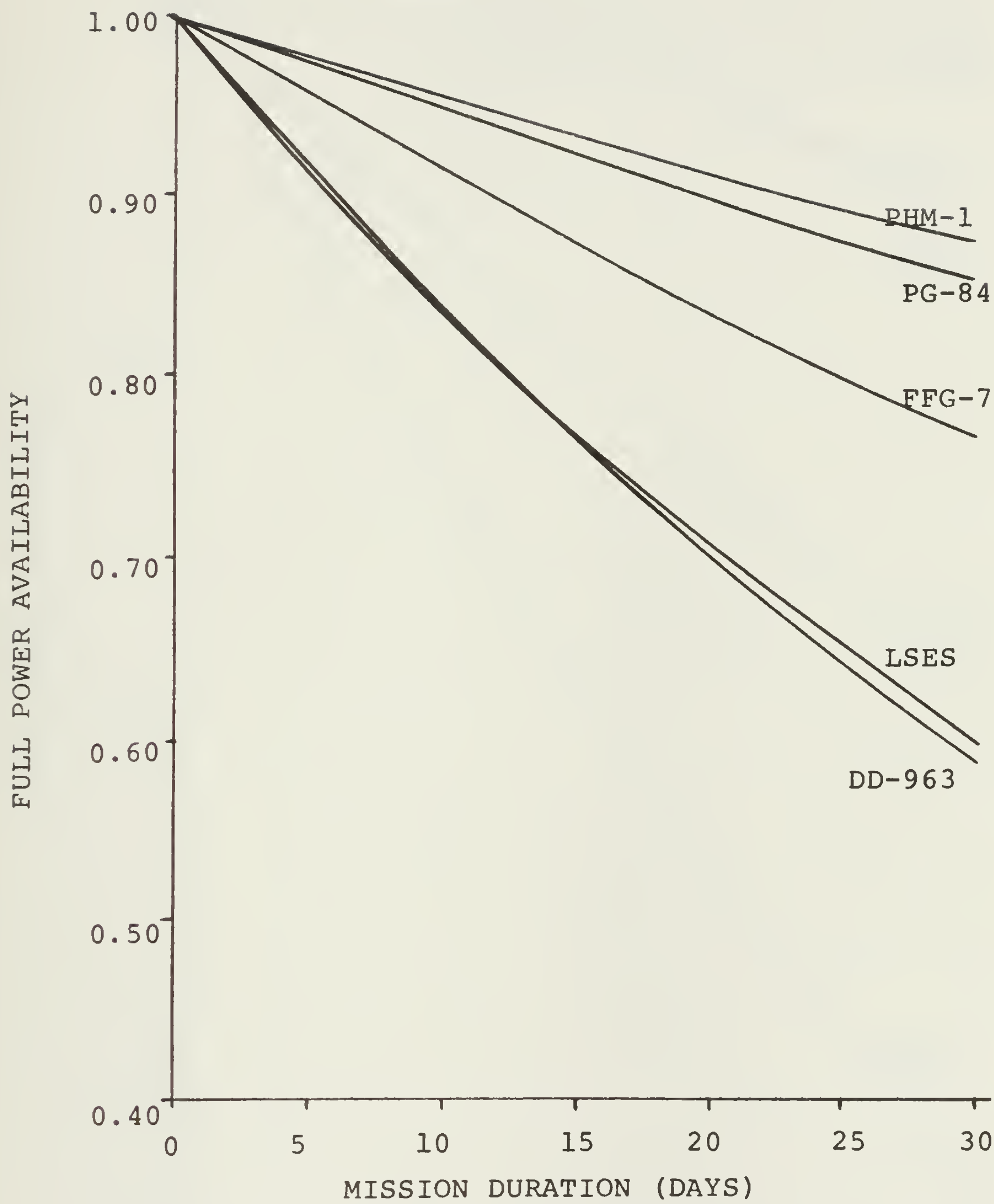


FIGURE 28 - FULL POWER AVAILABILITY VS. MISSION DURATION
PROPULSOR TYPE

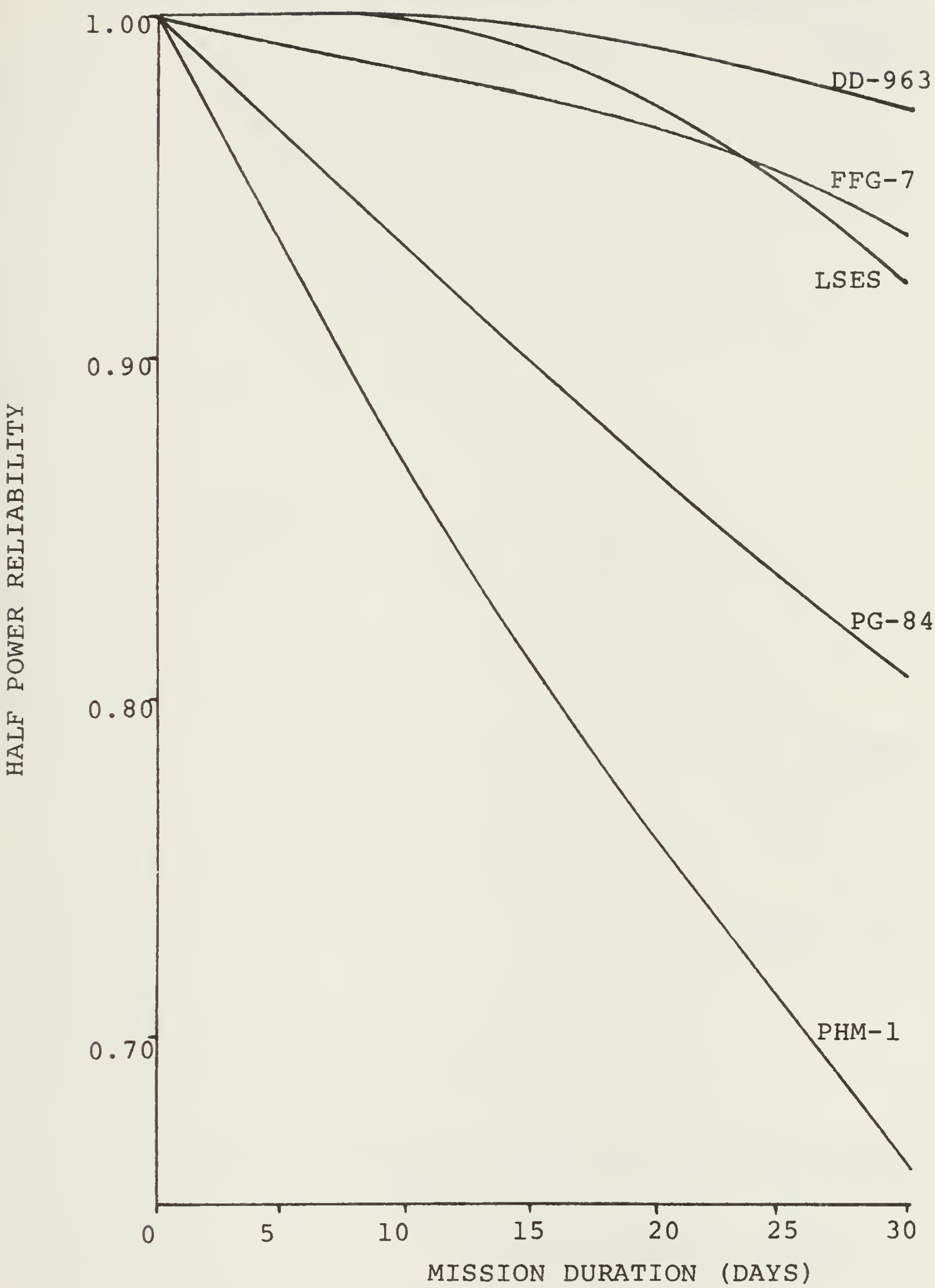


FIGURE 29 - HALF POWER RELIABILITY VS. MISSION DURATION
PROPULSOR TYPE

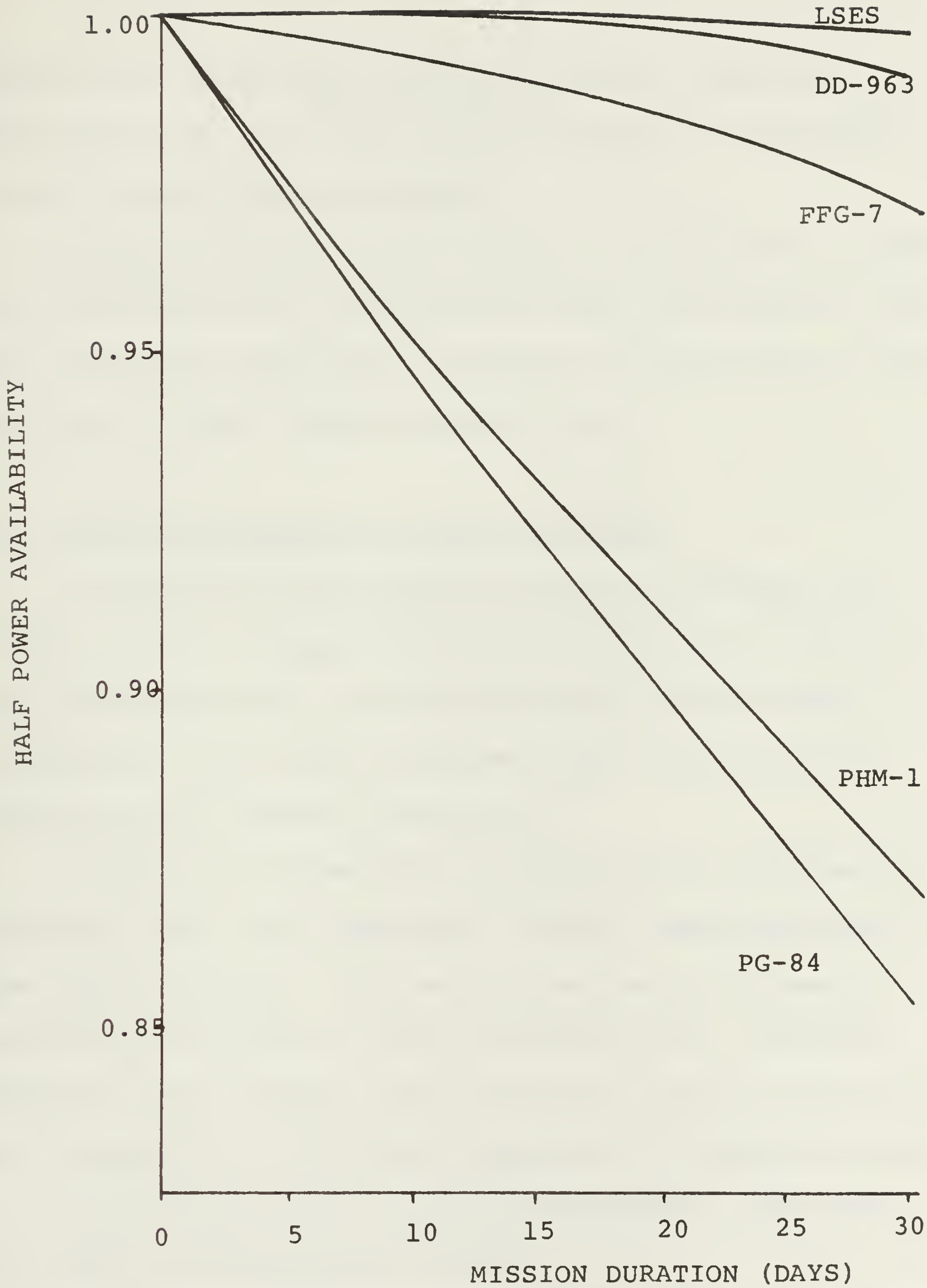


FIGURE 30 - HALF POWER AVAILABILITY VS. MISSION DURATION
PROPULSOR TYPE

reliability, while the effects of waterjet pumps and CRP propellers leave the relative standing of propulsion system availabilities unchanged.

Figure 31 presents the "cruise" reliabilities. With the type of propulsor incorporated into the analysis, the high performance propulsion systems are significantly less reliable in their cruise configuration.

6.5 R/M/A as Predicted by Ship Designers

Although high performance propulsion systems have similar levels of reliability and availability resulting from redundancy as do their conventional displacement counterparts, the use of waterjet pumps significantly degrades their overall reliability.

It has been assumed that except for the differences in propulsor type, all generically similar components have the same MTBF and MTTR. This may not be the case, however, since MTBF and MTTR can vary as pointed out in previous chapters. The designer can improve the overall reliability and availability of the high performance propulsion system by installing more reliable and maintainable components than used in conventional designs.

This analysis will investigate the propulsion system reliabilities and availabilities as predicted by the designers and evaluate the difference between the MTBF and MTTR data used.

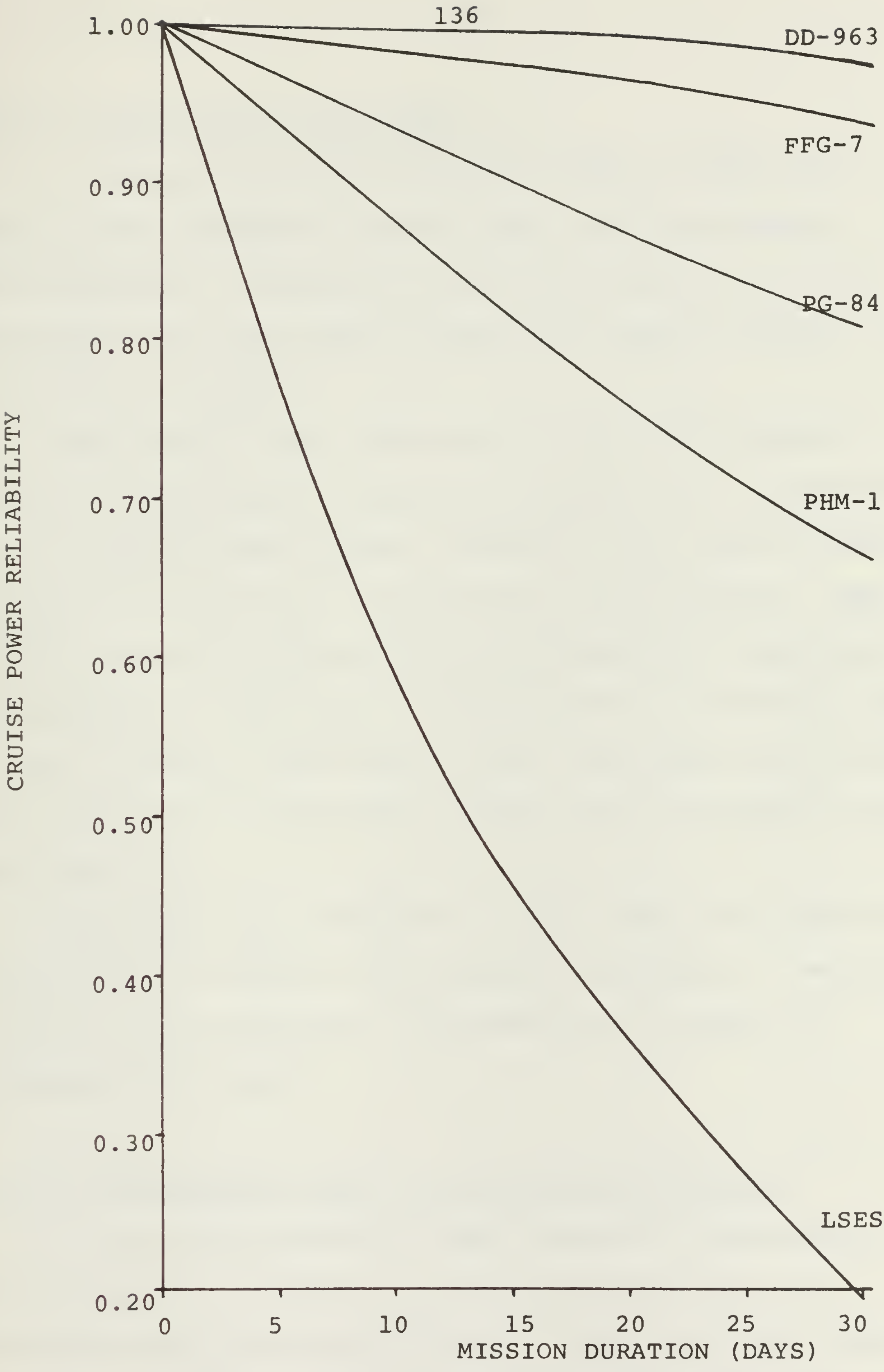


FIGURE 31 - CRUISE POWER RELIABILITY VS. MISSION DURATION
PROPULSOR TYPE

6.5.1 Reliability Predictions

Tables 15 through 18 list the MTBF and MTTR data used by the designers of PHM-1, LSES, FFG-7 and DD-963 respectively. No design data was available for the PG-84. Therefore, the same MTBF and MTTR data used in Section 6.4 is used here as well.

Reliabilities and availabilities for full and half power are presented in Figures 32 through 36. The designers of the DD-963 expect higher reliability from many of their components than do any of the other ship designers. This is the reason the reliability and availability show such an improvement at full power. The designers of the FFG-7 expect lower reliability from many of their components which explains the reduction in full power and half power reliability and availability.

There are many differences in the MTBF's and MTTR's for similar and in cases, even identical components. The next section will investigate the differences in MTBF's to determine the causes.

6.5.2 Variations of MTBF for Identical Components

The use of different MTBF values for identical or near identical components can bias the comparison of reliabilities and availabilities of these propulsion systems. As shown in earlier work, the differences in MTBF could be

TABLE 15

PHM-1 MTBF/MTTR DATA

Component	MTBF	MTTR	R(5)	R(15)	R(30)	A
GAS TURBINE	21,133	NR	0.9943	0.9831	0.9665	--
G.T. L.O. COOLER	71,428	2.0	0.9983	0.9950	0.9900	0.9999
SHAFTING	327,869	NR	0.9996	0.9989	0.9978	--
BEARING AND SEAL	10,511	6.0	0.9886	0.9663	0.9338	0.9994
THRUST REV	6,150	5.0	0.9807	0.9431	0.8895	0.9992
WATERJET PUMP	4,514	NR	0.9737	0.9233	0.8526	--
L.O. PUMP	21,800	3.65	0.9945	0.9836	0.9675	0.9998
L.O. COOLER	71,428	2.0	0.9983	0.9950	0.9900	0.9999
INLET DUCT	160,000	NR	0.9993	0.9978	0.9955	--
F.O.S.P.	61,199	0.8	0.9980	0.9941	0.9883	0.9999
CLUTCH	190,597	NR	0.9994	0.9981	0.9962	--
REDUCTION GR	6,456	NR	0.9816	0.9458	0.8945	--
DIESEL	3,595	13.0	0.9672	0.9047	0.8185	0.9964
F.O. STRAINER	32,512	0.3	0.9963	0.9890	0.9781	0.9999
L.O. STRAINER	32,512	0.3	0.9963	0.9890	0.9781	0.9999
F.O. HEATER	333,330	1.2	0.9996	0.9989	0.9978	0.9999

TABLE 16

LSES MTBF/MTTR DATA

Component	MTBF	MTTR	R(5)	R(15)	R(30)	A
GAS TURBINE	10,500	NR	0.9486	0.9663	0.9337	--
G.T. L.O. COOLER	90,000	4.5	0.9487	0.9960	0.9920	0.9999
COUPLING SHAFT	11,600	6.0	0.9897	0.9694	0.9398	0.9995
WATERJET PUMP & GEAR BOX	6,700	NR	0.9822	0.9477	0.8981	--
L.O. PUMP	21,800	3.0	0.9945	0.9836	0.9675	0.9998
L.O. COOLER	45,000	4.5	0.9973	0.9920	0.9841	0.9999
INLET DUCT	6,100	3.7	0.9805	0.9427	0.8887	0.9994
THRUST REVERSER	6,150	6.0	0.9807	0.9431	0.8895	0.9990
F.O.S.P.	3,760	4.5	0.9686	0.9087	0.8257	0.9988
F.O. HEATER	90,000	4.5	0.9987	0.9960	0.9920	0.9999
F.O. STRAINER	60,000	4.5	0.9980	0.9940	0.9881	0.9999
L.O. FILTER	30,000	4.5	0.9960	0.9881	0.9763	0.9998

TABLE 17

FFG-7 MTBF/MTTR DATA

Component	MTBF	MTTR	R(5)	R(15)	R(30)	A
F.O.S.P.	2,800	3.1	0.9580	0.8794	0.7733	0.9989
F.O. HEATER	14,800	3.8	0.9919	0.9760	0.9525	0.9997
F.O. STRAINER	10,000	4.0	0.9881	0.9646	0.9305	0.9996
LM2500	4,000	NR	0.9704	0.9139	0.8353	--
L.O. COOLER	90,500	3.0	0.9987	0.9960	0.9921	0.9999
CLUTCH	50,000	NR	0.9976	0.9928	0.9857	--
REDUCTION GEAR	200,000	NR	0.9994	0.9982	0.9964	--
L.O. STRAINER	60,000	3.0	0.9980	0.9940	0.9881	0.9999
L.O. PUMP	2,600	6.0	0.9549	0.8707	0.7581	0.9977
SHAFT & BEARINGS	200,000	NR	0.9994	0.9982	0.9964	--
CRP PUMPS & SERVO BOX	25,000	15	0.9952	0.9857	0.9716	0.9994
CRP HUB	125,000	NR	0.9990	0.9971	0.9943	--

TABLE 18

DD-963 MTBF/MTTR DATA

Component	MTBF	MTTR	R(5)	R(15)	R(30)	A
F.O.S.P.	40,000	6.2	0.9970	0.9910	0.9822	0.9998
F.O. HEATER	333,330	1.2	0.9996	0.9989	0.9978	0.9999
F.O. STRAINER	10,000	4.0	0.9881	0.9646	0.9305	0.9996
LM2500	22,600	NR	0.9947	0.9842	0.9686	--
L.O. COOLER	333,330	3.0	0.9996	0.9989	0.9978	0.9999
CLUTCH	100,000	NR	0.9988	0.9964	0.9928	--
REDUCTION GEAR	125,000	NR	0.9990	0.9971	0.9943	--
L.O. STRAINER	60,000	3.0	0.9980	0.9940	0.9881	0.9999
L.O. PUMP	40,000	5.8	0.9970	0.9910	0.9822	0.9998
SHAFT & BEARINGS	95,240	NR	0.9987	0.9962	0.9925	--
CRP PUMP & SERVO BOX	333,330	9.0	0.9996	0.9989	0.9978	0.9999
CRP HUB	1,000,000	NR	0.9998	0.9996	0.9993	--

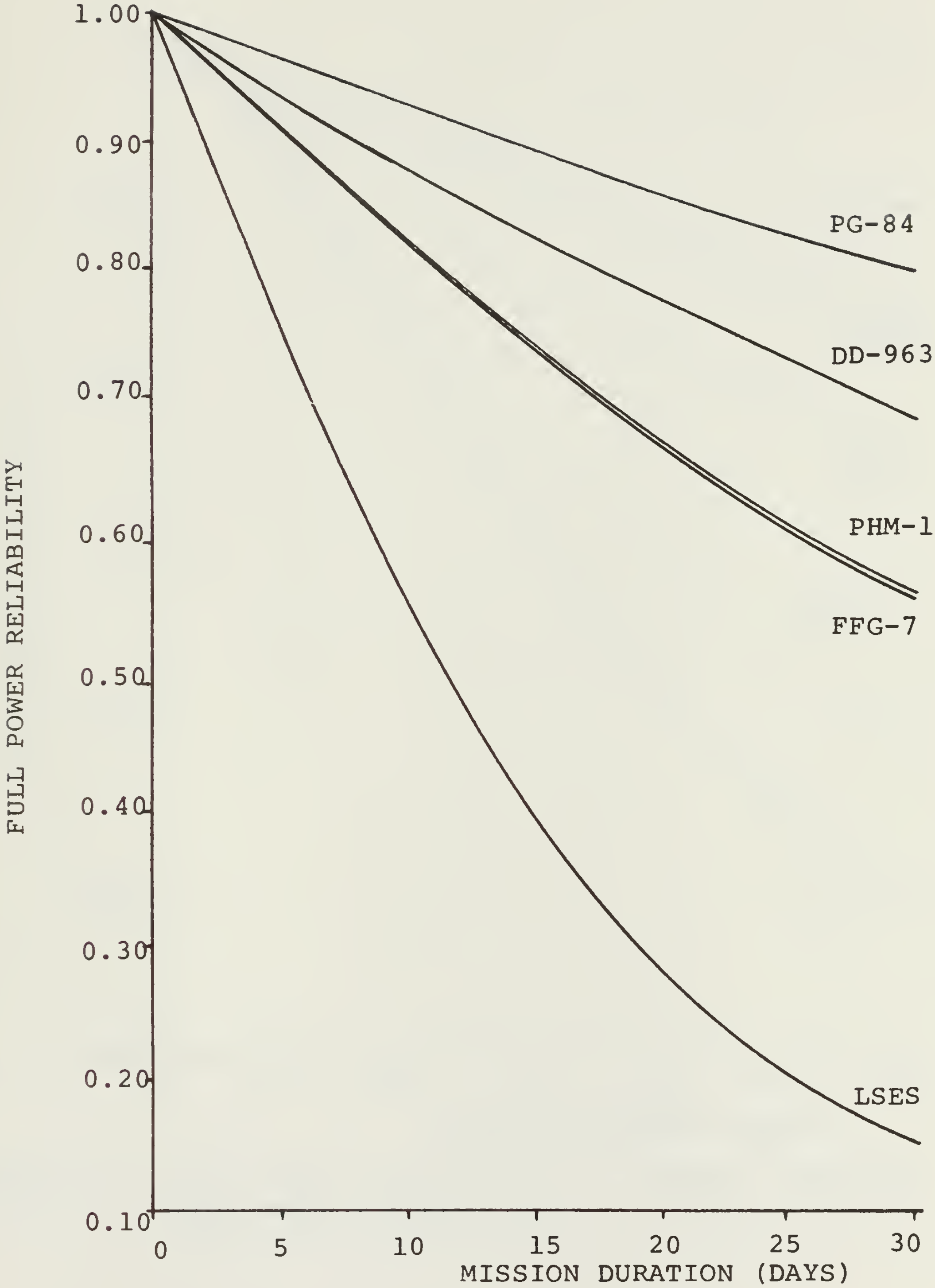


FIGURE 32 - FULL POWER RELIABILITY VS. MISSION DURATION USING SHIP PROJECT DATA

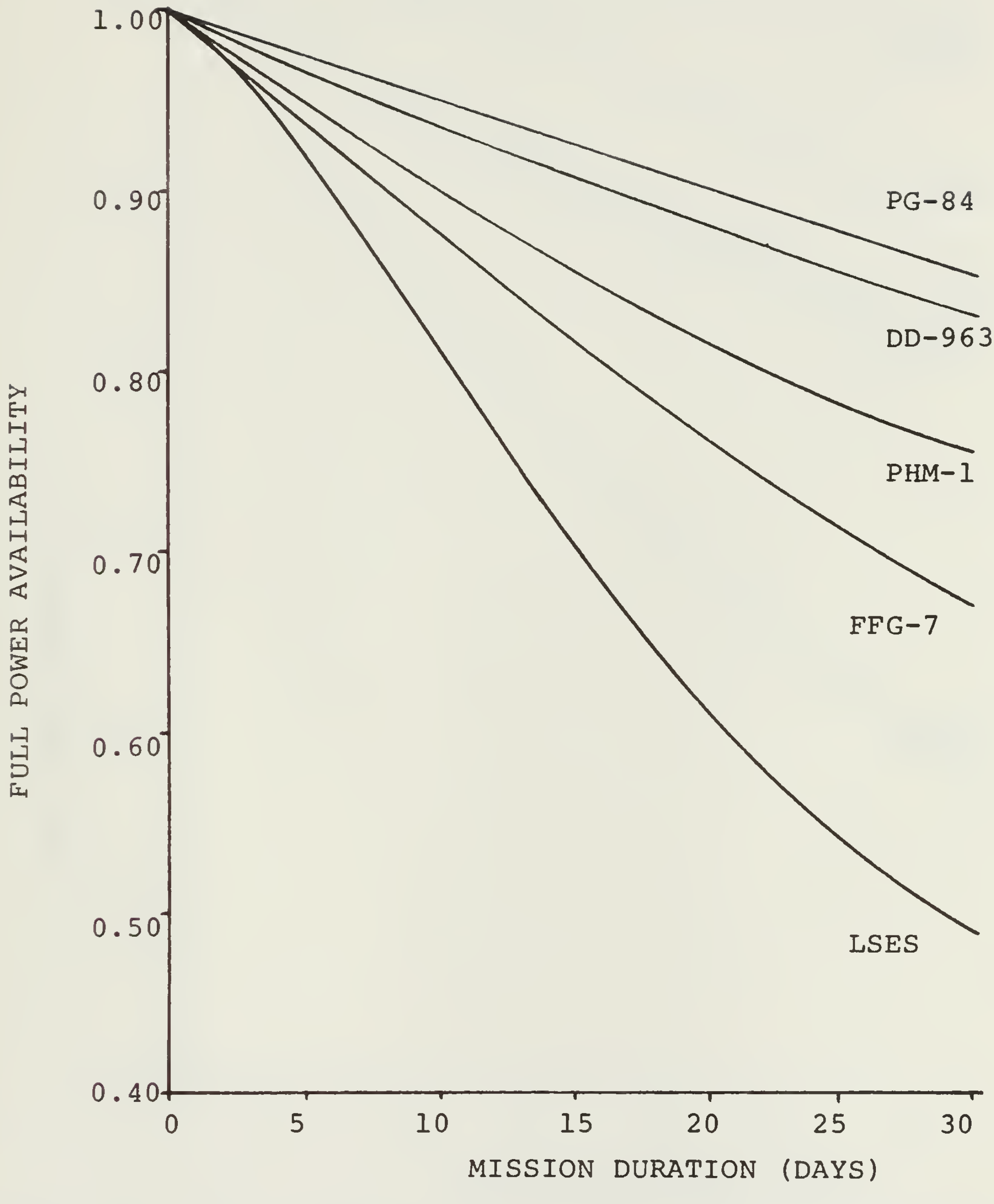


FIGURE 33 - FULL POWER AVAILABILITY VS. MISSION DURATION USING SHIP PROJECT DATA

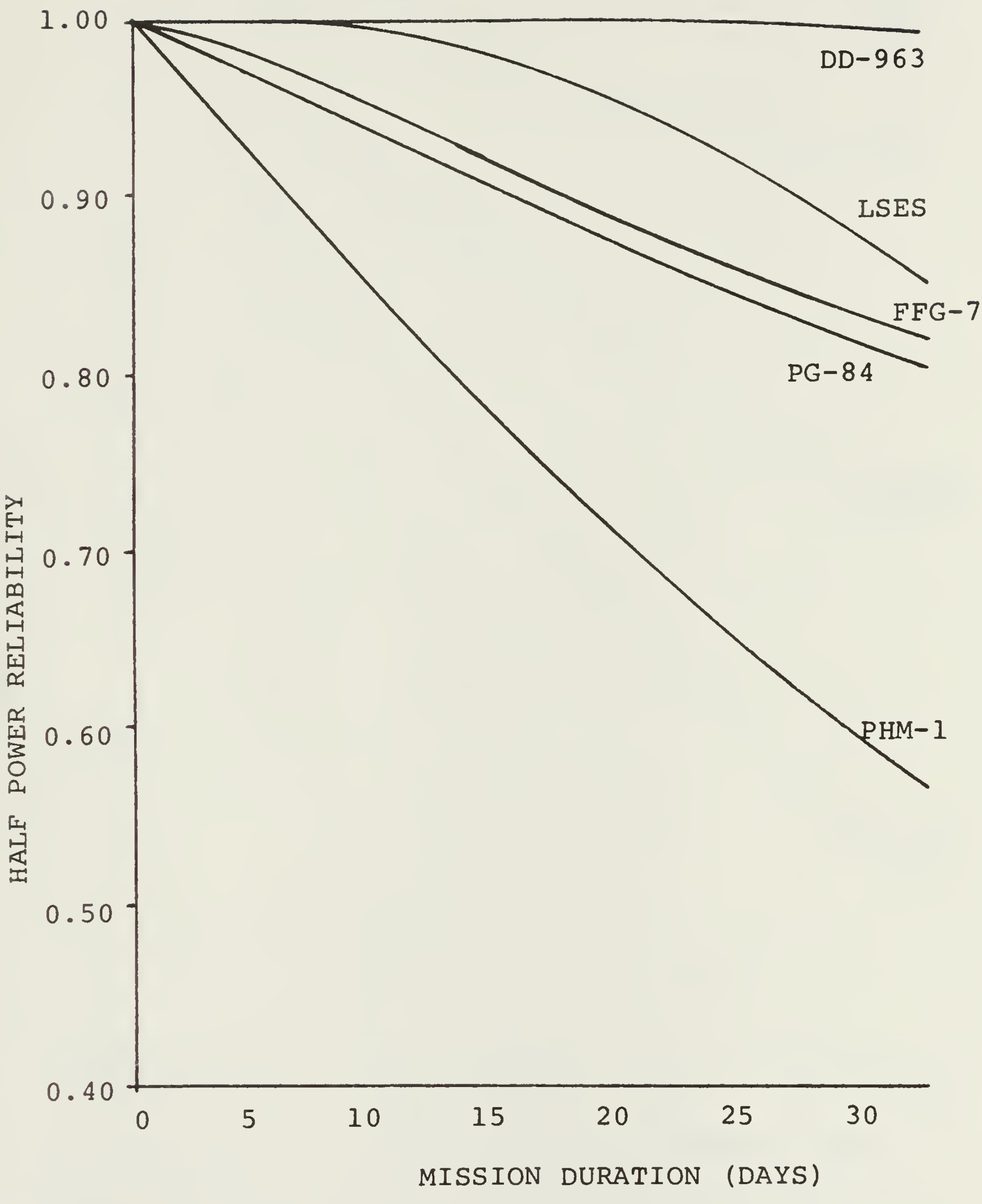


FIGURE 34 - HALF POWER RELIABILITY VS. MISSION DURATION USING SHIP PROJECT DATA

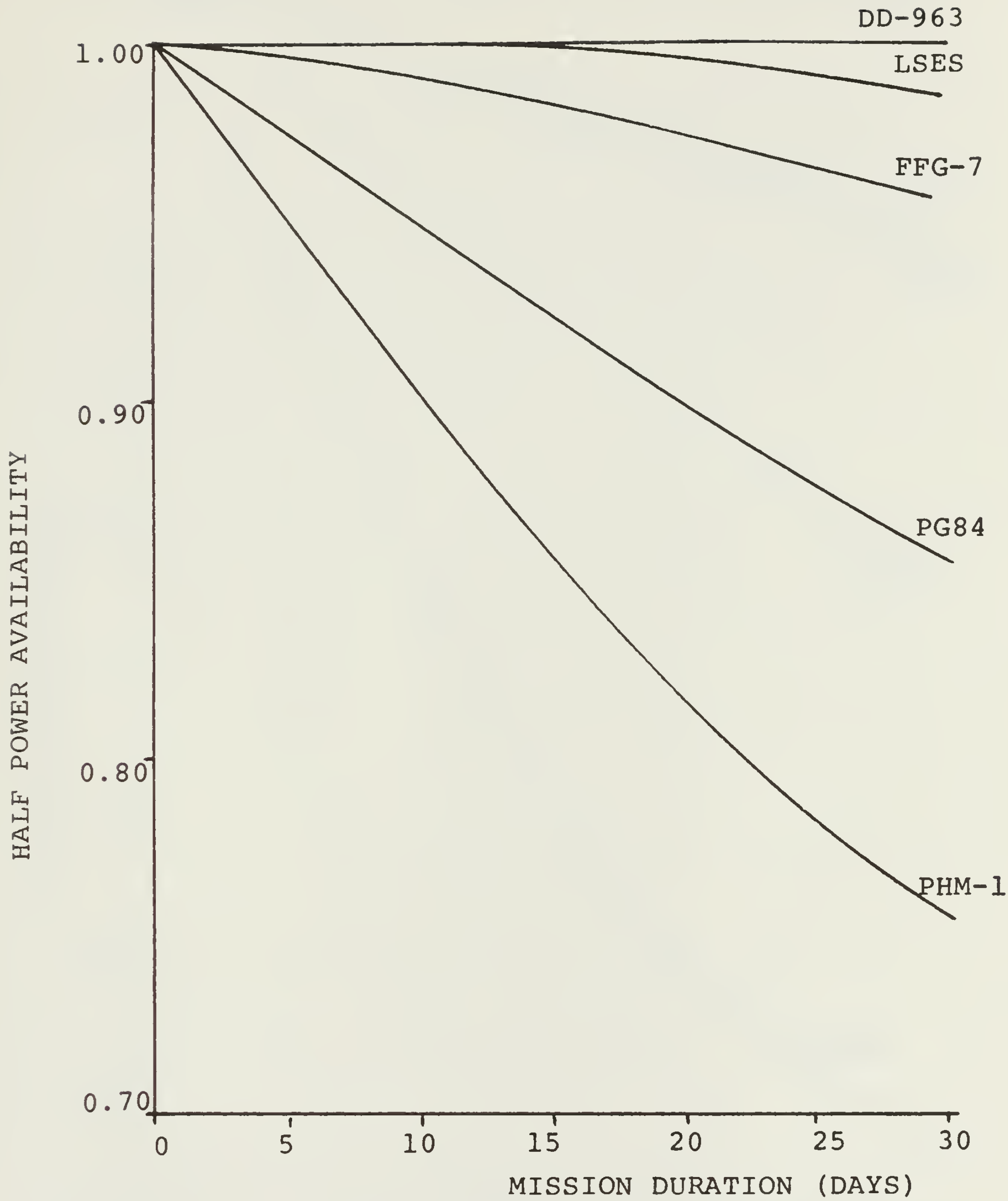


FIGURE 35 - HALF POWER AVAILABILITY VS. MISSION DURATION USING SHIP PROJECT DATA

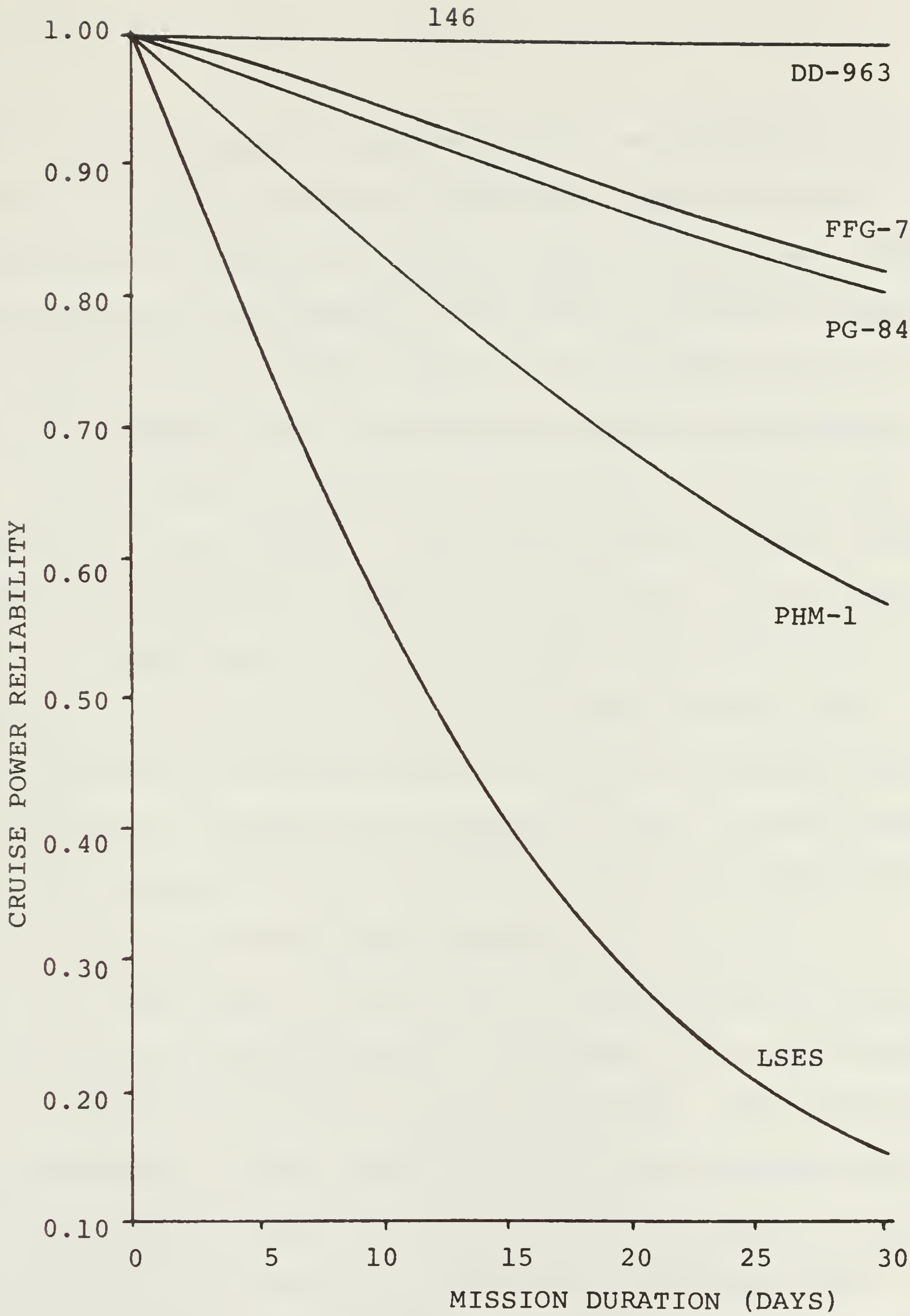


FIGURE 36 - CRUISE POWER RELIABILITY VS. MISSION DURATION USING SHIP PROJECT DATA

due to loading, design characteristics, environment, and service life. For identical components the only one of consequence is component loading, since it is assumed design characteristics, environment and service life are the same from ship to ship. Other reasons not discussed before are error in judgement and insufficient operating experience.

The one component which is identical on four of the five ships is the LM2500. By looking at the various values of MTBF in Table 19, a wide spread is noted. It is known that the PHM-1 rates its gas turbine at only 75% of max continuous power (16,000 SHP), while LSES, DD-963, and FFG-7 rate their gas turbines at 100% max continuous power (21,000 SHP). The PHM's gas turbine is more lightly loaded than the others, so it may be expected that its MTBF would be less. This is not the case however.

The time frame in which each of these ship reliability reports were produced provides the answer. The first naval application of the LM2500 was in the DD-963. The only R/M/A data available at this time was from the manufacturer and this was what was used. The PHM-1 project developed shortly after DD-963 and also had to use this preliminary data. This accounts for the high MTBF values used by these ships. The FFG-7 design didn't begin until several years later and with it a propulsion system land based test site was used to gain operational experience and develop reliability data.

With the land-based test site and DD-963 operating experience, the FFG-7 designers could make a better estimation of MTBF for the gas turbine. The FFG-7 MTBF is significantly lower than previous estimates, but is based on much more experience.

The LSES design is about three years behind the FFG-7 and again has much more experience to base an estimate of gas turbine MTBF on. The LSES designers have used a more optimistic value than FFG-7, perhaps feeling that continuing developments and improvements in the LM2500 justify better expected reliability.

The MTBF values for the reduction can be explained by differences in size and stress levels. The gear boxes of the high performance propulsion systems are very much smaller and operate at higher stress levels which reduce MTBF. The different values for the DD-963 and FFG-7 reduction gears cannot be explained by technical reasons since their MTBF's are opposed to those dictated by stress levels. The FFG-7 has both higher stress levels and MTBF. The same trend is noted with the CRP propellers. Again, the combination of this reduction gear with the CRP propeller in the DD-963 is a first time application at this large SHP rating. The lack of experience may have led the designers to use optimistic manufacturers' data. With actual operating experience from the DD-963, the FFG-7 designers were able to arrive at a more practical MTBF for both components.

TABLE 19

MTBF DATA SUMMARY

Component	PHM-1	PG-84	LSES	FFG-7	DD-963
LM2500/1500	21,133	6,450	10,500	4,000	22,600
CLUTCH	NA	50,000	NA	50,000	100,000
REDUCTION GEAR	6,456	200,000	6,700	200,000	125,000
SHAFT & BEARINGS	327,869/ 10,511	200,000	11,600	200,000	95,240
F.O. HEATER	333,330	333,330	90,000	14,800	333,330
F.O. STRAINER	32,512	60,000	60,000	10,000	10,000
L.O. PUMP	21,800	4,000	21,800	2,600	40,000
L.O. COOLER	71,428	90,576	45,000/ 90,000	90,500	333,330
L.O. STRAINER	32,512	60,000	60,000	60,000	60,000
INLET DUCT	160,000	NA	6,100	NA	NA
WATERJET PUMP	4,514	NA	6,700	NA	NA
THRUST REVERSER	6,150	NA	6,150	NA	NA
CRP PUMP	NA	100,000	NA	25,000	333,330
CRP SERVO BOX	NA	333,000	NA	25,000	333,330
CRP HUB	NA	100,000	NA	125,000	1000,000
DIESEL	3,595	3,000	NA	NA	NA

6.6 Summary and Conclusions

It has been shown that there is an equivalent degree of redundancy both in primemovers and support subsystems between both high performance and conventional displacement propulsion systems. Also, the full power reliability of any ship with high powering requirements could be improved if fewer higher powered primemovers were used. Thus there is a need for high power gas turbines.

In going to fewer high power gas turbines, the redundancy at half power is lost and reliability at this power level will be degraded. There is a definite trade-off to be made here by the designer, depending on his overall requirements and priorities.

The lighter weight and more compact gear boxes and waterjet pumps of the high performance propulsion systems are effective in reducing the propulsion systems impact, but their lower reliability and availability must be accepted.

High performance ships might be expected to operate at their conventional displacement counterparts, thus with the same components they will always be less reliable. They will have no primemover redundancy in this configuration while the conventional propulsion systems operating in a half power configuration will have half its primemovers in reserve.

In comparing the propulsion systems when evaluated using their own ship project failure rate and repair rate data, some size and loading variations have been noted, but the overriding influence seems to be the availability and source of MTBF and MTTR data at the time of each design. Thus, what may look like more reliable equipment is in fact only more reliable on paper as dictated by data used.

Overall, the high performance ship designers have had to install high power propulsion systems with a small weight and volume budget. They have been able to use about the same degree of redundancy in support subsystems of the propulsion system, but have had to use some highly stressed components. Thus, some reliability and availability has been sacrificed from the start. The need for configurations in excess of half power to take advantage of the reduced drag at high speeds, has forced these vessels to operate at their least reliable state a significant portion of the time.

The conclusion to be drawn from this is that in reducing the weight and volume impact of the propulsion system, the high performance ships have had to sacrifice a significant amount of reliability and availability.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

High performance propulsion systems have lower ship impact than conventional displacement propulsion systems. This low impact allows reassignment of weight and volume to improve payload carrying capability or some other performance features, such as speed or endurance.

One of the most important areas of operability as related to total ship performance is Reliability/Maintainability/Availability. R/M/A is influenced directly by system configuration, mean-time-between-failures, and mean-time-to-repair.

MTBF is influenced by component loading, design characteristics, service life, operating environment and the preventive maintenance performed. In current R/M/A predictions by naval ship designers, these influencing areas do not seem to be taken into consideration. No conclusive correlation between MTBF's and these five factors could be found; nor were any differences observed between high performance and convention propulsion component MTBF's that could be attributed to such influences.

MTTR is influenced by machinery accessability, spare parts, tools, and manpower availability, and shop capability. There are noticable differences in these influencing factors between high performance and conventional displacement

propulsion systems. The results of these differences indicate that high performance ships should have longer MTTR's. There is no indication that ship designers use higher MTTR's in the reliability and availability analyses of high performance propulsion system.

Configuration encompasses the degree of redundancy and type of components installed. The only differences in types of components between high performance and conventional displacement ships were the types of propulsors. There are no differences observed in the degree of propulsion support subsystems redundancy. The amount of installed shaft horsepower has more influence on propulsion system redundancy than does vehicle type.

Taking all of the above into consideration, high performance ships would appear to have lower reliability and availability than conventional displacement ships. However, there is not enough high performance ship operating experience in the fleet to substantiate this with service data.

Conventional displacement ships can be redesigned to reduce system impacts, but in so doing, there will be a loss in operability.

The following recommendations are advanced:

Conventional displacement designers should consider propulsion design concepts typical of high performance ships to take advantage of weight and volume savings.

High performance ship designers should take a more realistic look at the potential decrease in reliability and availability due to their present design philosophy.

The U.S. Navy should develop more rigorous reliability and availability methodology to take into account all the significant factors which influence MTBF and MTTR.

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